The
ENGINEERS' DIGEST
REVIEW OF WORLD ENGINEERING PROGRESS

VOL. X. No. 10 OCTOBER, 1949 4s. 6d

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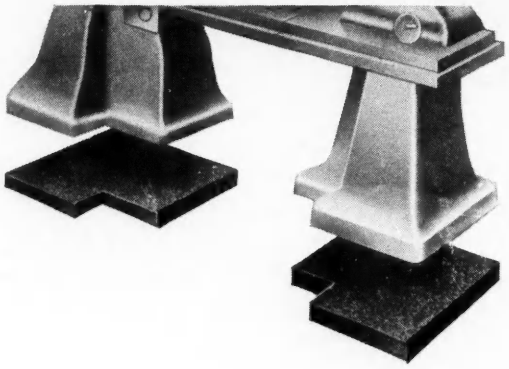

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IN THIS ISSUE

ONE of the basic problems of acoustics is that of determining the impression of noises of various loudness levels on the human ear. On the basis of experimental results so far obtained, G. G. Sacerdote has evolved, as explained in his article on page 346, two analytical expressions representing the acoustic sensation function, for low and high loudness levels. The expressions obtained are in close agreement with the standard sensation curve established from measurements by Fletcher and other investigators. An expression for the differential threshold, i.e., the increment of sensation which can just be detected by the ear, is also given.

* * *

The article "Vibrations in Power Transmission Lines" on page 355 indicates various formulae for the calculation of resonant frequencies, and gives the results of statistical observations carried out in Norway over long periods of time on the frequency of occurrence and amplitudes of vibration of transmission lines with copper and steel-cored aluminium conductors. Extensive observations were made to determine the effects of wind velocity, tensile loading, type of terrain, and daily and seasonal temperature variations. The uses of vibration dampers and the damping effect of insulator strings and cross-arms are also discussed.

* * *

The extension of materials loaded while submerged in liquids or in special atmospheres is not easily measurable. It is not always possible to attach the usual types of extensometer gauges to the test material or to obtain readings at easily accessible spots. The article on Optical Recording Extensometers on page 345 describes an arrangement used for the investigation of fibres under controlled atmospheres. By the use of rods and crossheads, the extension and load measurements are transmitted to the outside of the test chamber. Tension is exerted by a leaf spring, which also acts on a pivoted mirror turning with the spring deflection. A light beam is thus deflected and thrown onto a photographic plate moved automatically with the extension of the fibre. A load-deflection diagram appears on the photographic plate. The arrangement should, with suitable alterations, find a wider application than for research on fibres only.

A rapid-reversing drive for planing machines, developed by the Brown Boveri Company, is described in an article by M. Steinebrunner. The equipment incorporated the principles of the Ward-Leonard control system, and gives rapid and shock-free operation. Formulae for the calculation of the reversing time, and the total WD^2 of the moving parts, are given, as well as graphs based on experimental results and oscillograph recordings.

* * *

A new type of spring, known as the Neg'ator, and having a non-linear stiffness characteristic, is described on page 359. This spring is designed as a tight coil of flat material, and stiffness is varied along the spring length by varying the pre-stressing of the material during manufacture. The Neg'ator spring can be used in a wide variety of ways as a holding or transmission element, particularly in mechanical or electro-mechanical systems.

* * *

The use of liquid nitrogen for the cooling of metal parts to be assembled by contraction or shrink fitting is strongly advocated by C. Deschars in his article on page 363. The process is simple, rapid and economical, and has been successfully applied for various types of work, for instance for the fitting of cylinder liners, valve seat inserts, cylinder heads, bearing rings, and axle spindles, and for the reclamation of broken crankshafts of presses.

* * *

The development of welding and the resulting heavily increased use of structural steel and rolled sections have brought in their stride amazing improvements in the quality of cast iron products. The major professional institutions have set up special committees for research on cast iron, and a large number of high-grade cast irons of greater strength and ductility and special grain structure have been on the market for some time. A much wider use of these new materials could well lead to a substantial reduction of production costs in factories where designers are still adhering to the old notions of ordinary cast iron or mild steel. Composition and heat treatment have played a major rôle in this development. On page 348 we bring some information on research undertaken at the University of Aachen, Germany, during the war. This research has led to the development of rolled, pressed, or forged cast

Concluded on page 370.

Some Experiences with the Creep Behaviour of Materials

By A. JOHANSSON. (From *Teknisk Tidskrift*, Vol. 79, No. 8, February 19, 1949, pp. 127-132, 15 illustrations.)

REPORTS on the creep strength and high-temperature deformation of various materials have previously been based mainly on laboratory tests. Information obtained in practice under service conditions is still extremely scarce. Details are given in the following of the experience gained with high-temperature materials in steam turbines, gas turbines, and creep-testing machines.

STEAM TURBINES

Experience has shown that creep can occur in the blading, the turbine discs and the labyrinth glands. Creep in the blading is much less pronounced than in the labyrinth glands. The clearance between the rings is not appreciably affected, since the increases in diameter of adjacent rings are similar. The effect of creep in the labyrinth plates is more important, as it can increase the frictional resistance of the turbine discs.

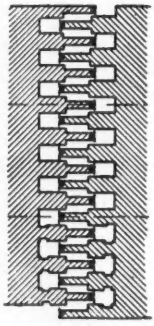


Fig. 1. **Labyrinth gland.**

Fig. 1 shows details of a labyrinth gland. If a radial clearance of 1·2 mm is provided between the projecting fin edges and the corresponding recesses, then, assuming that this clearance can be reduced to one-half of its value, a creep of 0·6 mm will be permissible. For radii of 400 mm this corresponds to a creep of 0·15 per cent. Assuming that the turbine is expected to operate for 100,000 hours, or approximately 11 years, it can, therefore, be concluded that the rate of creep should not exceed $1{\cdot}5 \times 10^{-6}$ per cent strain per hour. In general, the creep rate is well below this value, but considerable creep has been observed in certain cases. This has been attributed to two causes: (a) the abnormal creep behaviour of carbon steel, and (b) the effect of the operating temperature.

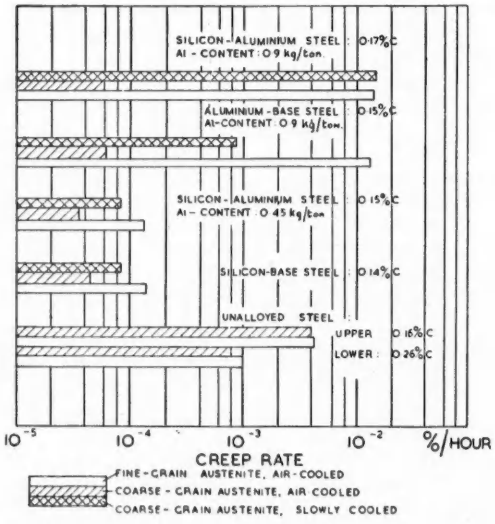


Fig. 2. **Effect of composition and structure on the creep strength of carbon steel after a 500-hour test at 10·55 kg/mm² and 455° C.**

Abnormal Creep of Carbon Steel.

Labyrinth plates of carbon steel can be used for temperatures up to 400-425° C. The creep strength of carbon steel can vary considerably, and according to American and British investigators, this may be due to insufficient deoxidation of the steel. Creep strength can be improved by an aluminium addition, which, however, must not exceed a certain percentage, in order to avoid detrimental effects. Special heat treatment and other alloying elements can also be used to increase the creep strength.

Fig. 2 shows various rates of creep obtained with different deoxidizing conditions and thermal treatments, for carbon steels with approximately 0·15 per cent carbon, tested at 455° C under a stress of 10·5 kg/mm². The lowest creep value, approximately 10^{-4} per cent strain per hour, is obtained with steel containing silicon or aluminium additions, the aluminium content being 0·45 kg/ton. When the aluminium content is raised to 0·9 kg/ton a marked increase in the creep rate is apparent, both in steels with an aluminium addition, and in silicon-aluminium steels. Excessive heat treatment increases the size of the intercrystalline cracks and counteracts the effect of the aluminium addition.

A sudden change in the creep rate of class A steels occurs when the aluminium content exceeds approximately 0·8 per cent. With other types of steels, the rise in the rate of creep can occur with a much lower percentage of aluminium. Ample evidence of this abnormal behaviour of carbon steels has been obtained in service. A 3000 kilowatt condensation turbine built by Stal in 1930 ran for several months with a steam temperature of 500-530° C. The blading and the labyrinth plates were made of ordinary carbon steel and creep was insignificant. On the other hand, considerable creep has been observed at working temperatures 100 degrees below this value. It has even been found that the labyrinth rotating on one side had considerable creep, while the creep in the other labyrinth was hardly measurable, although both labyrinths worked under similar conditions.

Effect of the Working Temperature on Creep.

In many cases it has been observed that the working temperature of a turbine plant exceeded the temperature originally foreseen, this being sometimes attempted in order to raise the efficiency of the installation. The fact is, however, that small variations of the working temperature can be the cause of considerable changes in the rate of creep.

The variation of the rate of creep with temperature is shown for carbon steels in Fig. 3. Even a 20° C temperature rise can have the very appreciable effect of increasing the creep rate to a ten times higher value. Although a 20° temperature rise is not necessarily dangerous in itself, in view of the abnormal creep characteristics of carbon steels, a temperature rise of this magnitude does signify that the material is approaching the critical creep zone.

As a result, some of the American manufacturers guarantee their machines solely on the understanding

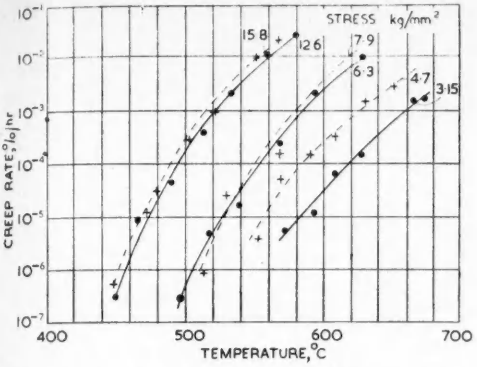


Fig. 3. **Variation of creep strength with temperature at constant stress values.**

that the following conditions are fulfilled:—(1) the turbine shall normally be run at 510° C (950° F); (2) it is not to be run for more than 5 per cent of its total operating time at a temperature of 524° C (975° F) ; and (3) it should not run for more than 1 per cent of its total operating time at a temperature of 538° C (1000° F). These conditions take account of the fact that in the temperature range considered, a temperature rise of 20° C during 5 + 1 per cent of the total operating time corresponds to an increase of 20 + 25 = 45 per cent of the total creep occurring during the total period of operation of the turbine.

Special Alloy Steels.

The present tendency is to run at increased temperatures, and carbon steels can, therefore, no longer fulfil the requirements. The Stal Company is, therefore, employing carbon-molybdenum steel as its standard material. For still higher temperatures, chrome-molybdenum steels are used.

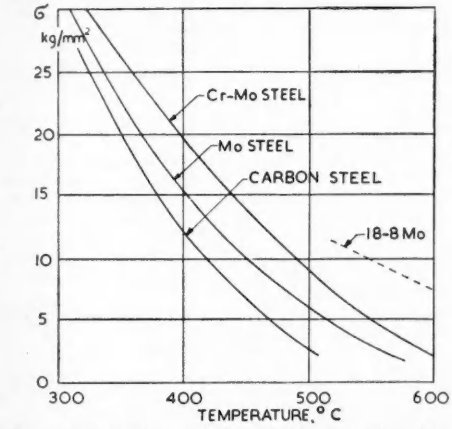


Fig. 4. **Steels for labyrinth plates. Stresses giving ultimate creep rates of 10^{-5} per cent per hour after a test of 1000 hours.**

Fig. 4 shows the relative creep strength of these steels, i.e., the maximum stress to which they may be subjected at a given temperature in order to have a creep rate not exceeding 10^{-5} per cent per hour after 1000 hours running. The curve for carbon steel refers to material with a normal creep characteristic. The difference between carbon steel and molybdenum steel is, therefore, generally greater than that shown in this diagram. The substitution of molybdenum steel for carbon steel enables a temperature rise of 50° C to be attained without altering the creep strength, and if chrome-molybdenum steel is used, the increase in temperature allowed is 100° C. When these high-grade steels are used in installations with a steam temperature of 500° C or even higher, it is necessary to have a temperature control which is effective within closer limits than those used for more normal temperatures. Both molybdenum and chrome-molybdenum steels are, furthermore, much less affected by variations in steel composition and heat treatment than carbon steel.

Both molybdenum steel and chrome-molybdenum steel are low-alloy steels. If higher-alloyed steels are also considered, for instance, austenitic steel of the 18-8 Mo type (17 per cent chromium, 12 per cent nickel, and 2·5 per cent molybdenum), or some of the

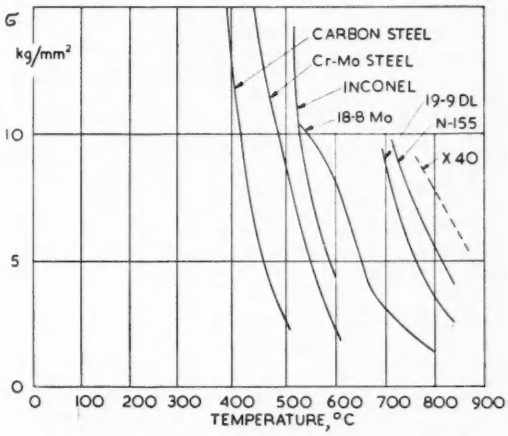


Fig. 5. **Stresses giving a creep rate of 10^{-5} per cent per hour after 1000 hours. Carbon steel, Cr-Mo steel, Inconel, 18-8 Mo, 19-9 DL, N-155, and X-40.**

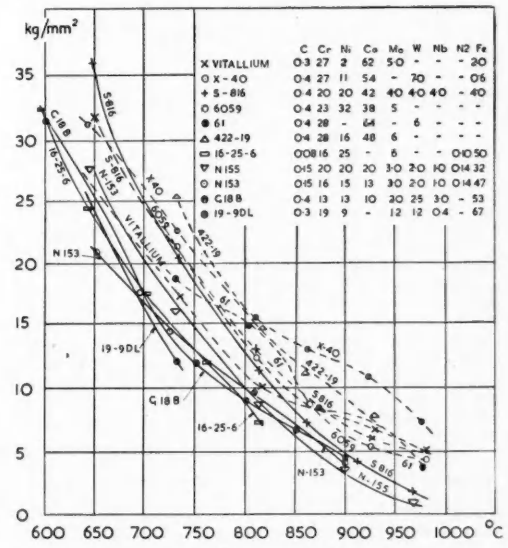


Fig. 6. **Stress to produce fracture after 1000 hours running, for various gas turbine materials.**

new gas turbine materials, then still higher temperatures can be reached (Figs. 4 and 6). In the United States, several steam turbine installations have been developed for a steam temperature of 565°C (1050° F), using materials with sufficient creep strength at this temperature.

It may be added that creep has also been observed in stationary parts, particularly in flange joints in the inlet section of the turbine. Even in the stationary part of the labyrinth gland, a certain amount of creep has been measured.

GAS TURBINES

The maximum temperature of the gas turbine is determined by the creep strength of the material used for the combustion chambers, the blading, and the turbine discs. The requirements for stationary turbines, which are designed to run for long periods, are different from those of a turbo-jet engine, which has a comparatively short life. In both cases, however, similar materials are employed, and the limiting considerations are based on the maximum permissible creep at operating temperatures. In general, the stationary gas turbine works in a lower temperature range, and, therefore, its components are less apt to reach elongations causing embrittlement and fracture of the material.

TABLE I. HIGH-TEMPERATURE MATERIALS FOR GAS TURBINES

Type	C	Cr	Ni	Co	Mo	W	Nb	Ti	Al	Use
19-9 DL ..	0·30	19	9	—	1·25	1·25	0·4	0·3	—	turbine discs
N-155 ..	0·15	20	20	20	3	2	1	—	—	
S-816 ..	0·40	20	20	42	4	4	4	—	—	blades
Nimonic 80	0·03	20	75	—	—	—	—	2	0·55	plates
Inconel ..	0·05	14	75	—	—	—	—	2·5	0·6	
254-E ..	0·18	25	23	—	—	—	—	—	—	
Stainless 17 : austenitic Cr-Ni steel with W and Nb										turbine blades, discs
BFS-7 : Cr-Ni-Co steel with Mo, W and Nb										

Table I gives particulars of the high-temperature materials used in turbines built by the Stal Company, and supplied by American, British and Swedish iron works. Many interesting test results have been obtained. Fig. 5 indicates the permissible stresses for some of these materials at various temperatures, assuming that the rate of creep is not to exceed 10^{-5} per cent per hour. Comparing, for instance, the N-155 steel and the low-alloy chrome-molybdenum steel, it is seen that the N-155 steel can withstand temperatures which are 200-250 degrees higher.

As a figure of merit for materials for jet engines, one frequently gives the limiting stress at which rupture occurs after 1000 hours' running. Fig. 6 gives this critical stress for some special materials, at various temperatures. The dotted lines denote cast steel materials with which some of the best values have been obtained; cast steels, however, suffer from the disadvantage of being more brittle than wrought steel materials. In practice, the latter property is decisive for the suitability of the material, and, therefore, the superiority of the cast steel materials, for the time being, is to some extent only an apparent advantage.

CREEP TESTING MACHINES

Creep tests at high temperatures have been carried out by the Stal Company since 1926, mainly on machines developed by Stal, of the type shown schematically in Fig. 7. The machine is so designed that it is free from tie-rods conducting heat out of the furnace, all components which transmit the load to the test-piece working in compression only, while the test specimen is loaded in tension. Ceramic materials are used to transmit the load from the external loading members to the test specimen, and the entire arrangement is insulated with ceramic materials to minimize heat losses and to obtain a uniform temperature during the tests. The machines serve for creep tests at temperatures up to 1000° C, and about 30 of these machines are employed both by the Stal Company and other Swedish firms. The holding members are of steel with 23 per cent chromium and 24 per cent nickel, one of the best high-temperature materials available when the first testing machines were originally designed, and only sustain a fraction of the stresses in the test specimens. Three columns transmit the load to the two clamping plates of the specimen, and are loaded in compression. The columns are of a large diameter in order to reduce creep stresses and to avoid buckling under the eccentric loading.

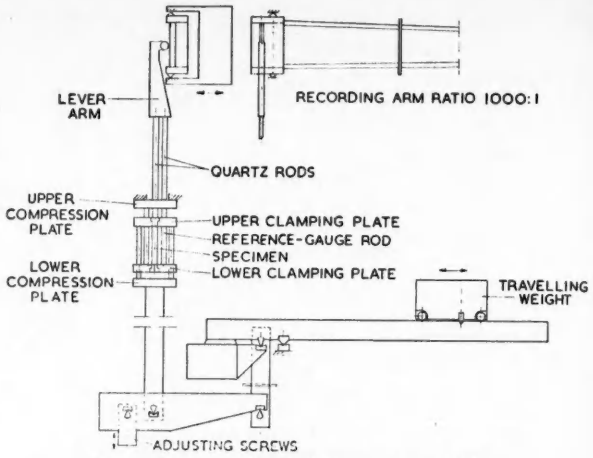


Fig. 7. Schematic sketch of creep testing machine.

DISCUSSION

The article given above is based on a lecture given in the Mechanical Section of the Swedish Technological Association. It was followed by a discussion with the following contribution by a mining engineer, G. LINDH.

A number of Swedish steel works are actively interested in the development of high-temperature alloys. The purpose of this contribution is to give some of the results achieved by Bofors AB in co-operation with Sandvikens Ironworks and the Swedish Air Force Research Station at Ulvsunda.

Professor Hultgren indicated various lines of research which may lead to the development of high creep strength materials, and I agree with director Grabe that precipitation-hardened alloys are among the most interesting materials, and are particularly suitable for wrought products. The Bofors-Sandvikens group is interested in wrought materials, although cast alloys are also being investigated.

An example of precipitation hardening is given by the wrought Bofors chrome-nickel-cobalt base " steel," with molybdenum, tungsten and niobium as precipitation-hardening elements. The iron content of this alloy is only 33 per cent. The material is heat treated at 1250° C for one hour and the structure is then altered into a homogeneous mass. Subsequent reheating (or ageing) at 800° C for about ten hours pro-

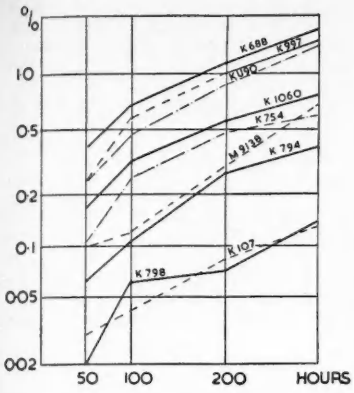
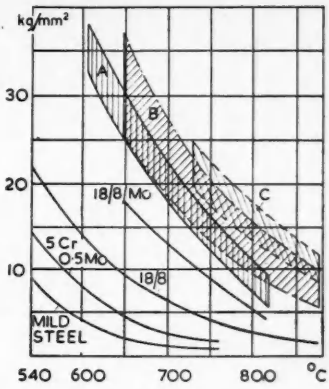


Fig. 8 (left).

Creep tests of Bofors ROT materials at 700° C, under a stress of 10 kg/mm².

Fig. 9 (right).

Stress to fracture values for fracture after 1000 hours. "A" type alloys are materials subjected to hot-cold-working; "B" alloys are precipitation-hardened alloys; "C" alloys are cast steel alloys (according to C. T. Evans, Jr.).



duces the precipitation effect and gives the fine grain distribution. The material is capable of sustaining operating temperatures of 650-750° C.

Both Bofors and Sandviken have a number of testing machines for long-time tests and creep tests. Bofors has six units for various types of creep tests, three larger machines for the precision measurement of creep, and two Losenhausenwerk creep-testing machines with automatic recording of creep according to the German specification DVM A-117/118, used for short-time tests conducted over a period of less than two days. The last-mentioned machines can also be used for creep fracture tests.

The laboratories also have various high-frequency units for specimens subjected to 50 kg loads, and have developed for customers a number of machines with 4-ton loads using 18-8 type materials with molybdenum, tungsten, titanium and niobium additions. The units undergo a 300 hours test, during which the stress is determined which gives a permanent strain of 1 per cent. The specimens are tested at constant temperature and constant load, and taken out after 50, 100, 200 and 300 hours for elongation measurements. The steel, designated Bofors ROT, is cold-drawn to obtain a high heat resistance. In America, this hot-cold-working process is carried out at high temperatures in the range of 650-750° C. The treatment takes place at recrystallisation temperature and is followed by a reheating or ageing process, the precipitation effect occurring while the steel is in a stressed condition. Fig. 8 gives the results of 300-hour tests of 9 different specimens of ROT materials at 700° C with a stress of 300 kg/mm². The curves differ considerably, although the specimens were of fairly similar materials. Brittleness can be caused by non-uniform reduction in cold-drawing, differences in grain structure, and variations in the precipitation-hardening effect. American in-

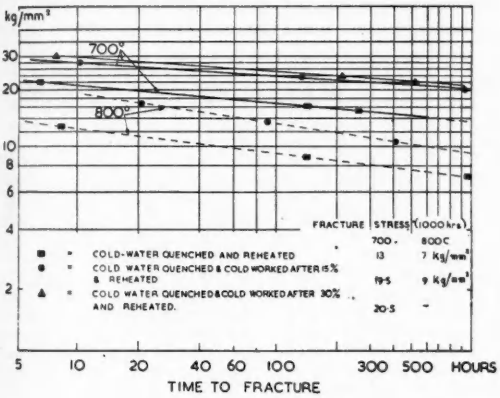


Fig. 10. **Creep fracture tests of BFS-6.**

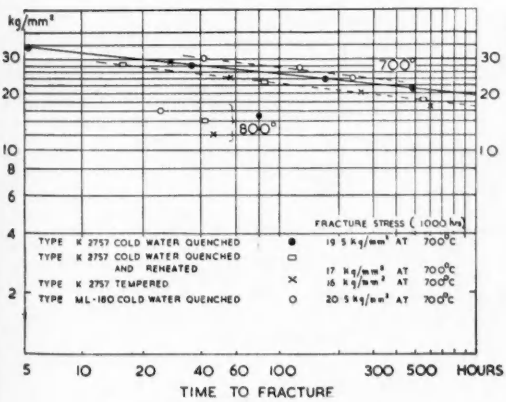


Fig. 11. **Creep fracture tests of BFS-7.**

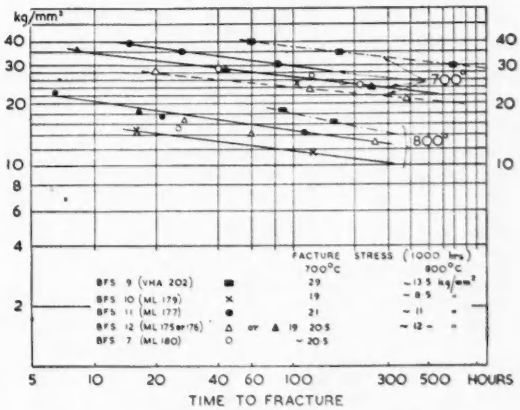


Fig. 12. **Creep fracture tests of BFS-9, 10, 11 and 12.**

vestigators also report difficulties in obtaining uniform creep characteristics for steels which are of a similar composition.

Fig. 9 gives further data on the heat-resistant steels developed during the war, and is based on the work of C. T. Evans, Jr. Figs. 11 and 12 are diagrams showing the stresses corresponding to fracture after 1000 hours at 700 and 800° C. Bofors and Sandvikens have also developed a series of experimental materials for loads of 50-150 kg from various cast alloy types. The alloy BFS-6 belongs to group A according to Fig. 9 and is an austenitic chrome-nickel-molybdenum steel improved by cold-working. As shown in Fig. 10, it has a creep strength (for 1000 hours to fracture) which changes from 13·19·5 kg/mm² at 700° C to 7-9 kg/mm² at 800° C. Type BFS-7, the properties of which are shown in Fig. 11, is a group B steel according to Evans's classification. This is a chrome-nickel-cobalt alloy with other precipitation additions. Its best value is approximately 20 kg/mm² at 700° C for a creep fracture time of 1000 hours. Finally, Fig. 12 shows the properties of BFS-9, a precipitation-hardened alloy which has a creep strength after 1000 hours of 29 kg/mm² at 700° C and 13·5 kg/mm² at 800° C. These values are higher than those of the group B materials. Type BFS-10 is a low-alloy steel, as compared with the other materials, and is also the least expensive type, but it still has good characteristics at 700° C. BFS-11 and BFS-12 are high-alloy materials. In conclusion, it can be said that the aim of the Swedish industry is to produce suitable creep-resistant materials for both jet-engines and stationary gas turbines.

RUSSIA

Thermocouple for the Measurement of Steel Bath Temperatures

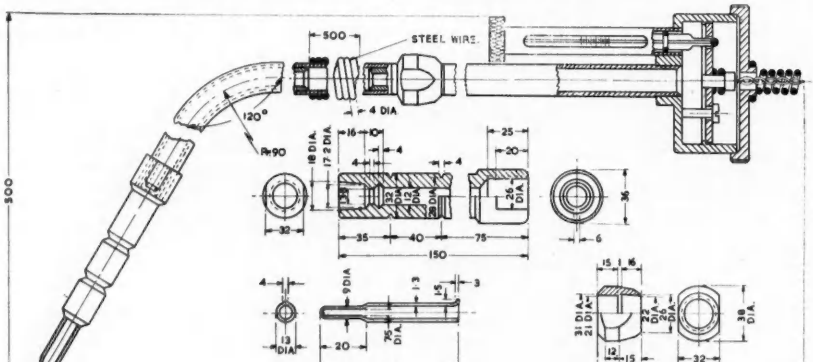
By V. G. Gruzin. (From *Zavodskaya Laboratoria*, No. 11, 1948, pp. 1396-7, 1 illustration.).

The thermocouple, comprising also its housing and operating handle, has been developed to take temperatures throughout the entire process of steel manufacture, from the steel furnace to the casting mould.

Platinum and platinum-rhodium wires of 0·5 mm diameter and 3 metres length are used, and must be annealed at 1000-1200° C for two hours; following this, they must be allowed to cool off for 3 minutes. The annealing is done by passing a current through the thermocouple. After the hot junction is welded and the couple calibrated together with its potentiometer, it is placed into the steel casing shown in the illustration. The hot junction is protected by a quartz sleeve closed at one end and drawn on an oxygen burner. The drawing is necessary to reduce the heat inertia of the sleeve and, thus, the time required to obtain a reading. The other end of the sleeve fits into the conical end of the steel casing and is covered by asbestos packing. The changing of the quartz sleeve does not take more than 10 seconds. This is important, as the sleeve must be changed after each reading. The steel casing is screwed to a mild steel pipe which is strengthened by a second pipe inserted therein, and further stiffened by steel wire wound around it, both wire ends being welded to the pipe. The end of the pipe remote from the steel casing is threaded and slotted. A clamping nut screws on to the thread and locates the position of a telescopic tube, on to which fits the end of a box enveloping the cold junction. The box also contains a thermometer, the casing of which, together with the telescopic tube, constitutes a handle for manipulating the whole assembly.

Normally, only the quartz sleeve is submerged in the steel bath, but it is quite safe to submerge the steel casing, as, by the time it reaches the molten metal, it will have passed through the layer of slag and will be covered by it. If there is no layer of slag, however, submerging of the steel casing is not recommended.

The time required for a reading is of the order of 20 to 45 seconds. After 40 to 80 readings, the hot junction must be renewed, and this is done by cutting approximately 8 mm off its end and re-welding. This necessitates the adjustment of the telescopic tube, as the length of the wires decreases. When the telescopic tube reaches the end of its travel, new wires approximately 1 metre in length, are welded on at the hot end, and the assembly is re-calibrated. When repeated readings are taken, the temperature of the cold junction rises, introducing an error which may reach 1 per cent. This is inadmissible, and, therefore, thermometer readings are recorded for each measurement so that the necessary correction can be made.



Thermocouple assembly. (Dimensions in millimetres).

Optical Recording Extensometer

By M. BASSIÈRE. (From *La Recherche Aéronautique*, No. 9, May/June, 1949, pp. 55-56, 4 illustrations).

FOR the measurement of the elongation of natural or synthetic fibres in controlled atmospheres or under liquids and at various temperatures, the following extensometer, shown schematically in Fig. 1, has been built. The fibre (1) is fastened, at its upper end, to a movable rod (2) which extends the fibre at a constant adjustable rate. The lower fibre end is fastened to a crosshead (3) which transmits the tension to a cantilever leaf spring (4), the deflection of which is measured optically. The stress-strain curve is registered on a photographic plate, which is moved vertically by the strain mechanism, the horizontal displacement of the light spot being proportional to the tension force.

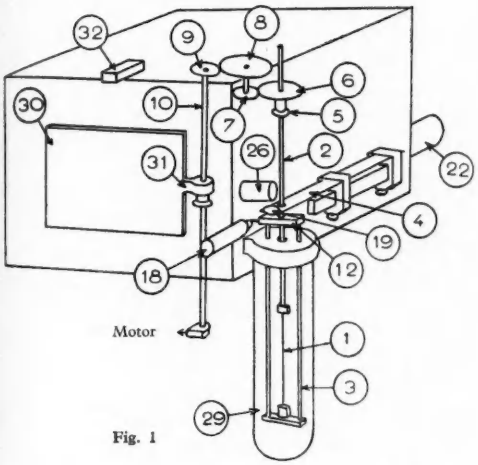


Fig. 1

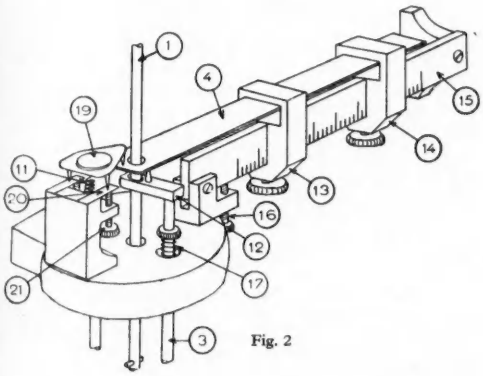


Fig. 2

A variable-speed motor with accurately controlled speed drives spindle (10) through a reduction gear (not shown). Spindle (10) is connected to nut (5) by gear train (9)-(8)-(7)-(6). Rod (2) is threaded at its upper end and displaced vertically as nut (5) turns with gear (6). Spindle (10) is also threaded and carries nut (31) which displaces the photographic plate (30) in a vertical direction as the spindle (10) turns.* For easier assembly and change of specimen, nuts (5) and (31) are split. The speed range is such that the rate of fibre extension can be varied from 0·02 to 15 mm per minute. Gears (9) and (8) are interchangeable so that the ratio of the rate of fibre extension to the speed of the photographic plate can be varied from 1 to 25. No play is allowed between the movements of rod (2) and plate (30).

The fibre is attached to screwed jaws fixed to the ends of rod (2) and crosshead (3). The crosshead can be placed in a casing (29) which may be filled with liquid or gases and thermostatically controlled. The two parallel rods of the crosshead are joined at the top by a cross-bar (12). This transmits the fibre tension to leaf spring (4), which is fixed to the frame and the deflection of which is measured. The spring deflection for a given fibre tension can be changed either by interchanging leaf springs of different thicknesses, or by adjusting the spring supports (13) and (14) so that the effective length of the spring is varied (Fig. 2). A graduated scale indicates the actual effective length of the spring. The maximum fibre tension force, corresponding to full-scale deflection of the light spot on the photographic plate, can thus be varied from 40 to 1500 grams. The weight of the crosshead itself is balanced by springs (17). Screw (16) serves to adjust the height of the assembly and controls the initially horizontal position of mirror (19). The displacement of cross-bar (12) can be observed through microscope (18) or recorded by the optical device. Mirror (19) is mounted on a three-point support. Two legs of the support rest on blades (20) which can be adjusted by screws (21) and determine the axis of rotation of the mirror. The third point rests on the extremity of spring (4). A small tension spring (11) holds the mirror to the points of support. The angle of rotation of the mirror, therefore, transmits the spring deflection and, hence, the fibre tension. As all displacements are very

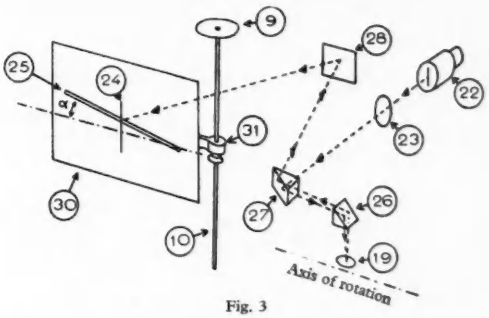


Fig. 3

small, the transmission ratio is practically linear over the whole range. Two stops (not shown) prevent any sudden displacement of the spring which might upset the apparatus.

A narrow vertical beam of light produced by lamp (22) (Fig. 3), is focussed by lens (23), directed onto the mirror (19) by prisms (26) and (27), reflected through the same prisms onto another mirror (28), and projected onto a narrow adjustable transverse slit (25) which covers the photographic plate. Only a point of light is thus permitted to impinge upon the photographic plate. As tension is put on the fibre, the spring (4) deflects and tilts mirror (19) so that the vertical beam of light is displaced horizontally across the transverse slit (25). However, as spring (4) deflects, the crosshead

*Switch (32) automatically stops the motor when plate (30) has travelled its own height.

(3) is also slightly displaced. If the true fibre elongation is to appear on the photographic plate, the displacement of crosshead (3) has to be deducted from the displacement of rod (2) or, to some chosen scale, from the displacement of the photographic plate itself. Since the displacement of crosshead (3) is proportional to the tension force tilting the mirror, this correction is made automatically if slit (25) is inclined at a suitable angle α to the horizontal. This angle α obviously depends on the scale chosen for the vertical movement of the photographic plate, i.e., on the choice of gear ratio (9)-(8). It must, therefore, be adjustable.

ITALY

Calculation of the Function of Acoustic Sensation

By G. G. SACERDOTE. (From *Alta Frequenza*, Vol. 43, No. 1, February, 1949, pp. 17-23, 5 illustrations).

THE fundamental problem of psychophysical acoustics is to determine the relationship between a quantity representing the "acoustic sensation," and the physical excitation by which it is produced.

We shall assume that two experiments have been carried out so as to determine the values

$$S_1 = f(L_1) \quad \text{and} \quad S_2 = f(L_2)$$

for the case $S_1 = 2S_2$. In these equations, S = sensation, and L = loudness level in decibels. We then have

$$f(L_1) = 2f(L_2).$$

Therefore, there exists a relationship

$$L_1 = g(L_2).$$

If g is known, f can be determined by solving the functional equation

$$f[g(L)] = 2f(L) \quad .. \quad .. \quad (1)$$

It is found experimentally that, over a considerable range of the audible spectrum, L_1 and L_2 bear a linear relation to each other. We can thus write

$$g(L) = aL + b,$$

so that eq. (1) becomes

$$f(aL + b) = 2f(L) \quad .. \quad (1')$$

The solution of eq. (1′) is obtained by substituting the product of two functions

$$f(L) = F(L) \times (mL + n)^r \quad .. \quad (2)$$

Then, putting $m = 1$, $n = b/(a-1)$, and $r = \log 2/\log a$ (logarithms to the base 10), we have

$$(mL + n)^r = \left(L + \frac{b}{a-1}\right)^{\frac{\log 2}{\log a}}$$

which is a particular solution of eq. (1′). Substituting the expressions of eqs. (2) and (3) in eq. (1), we find that $F(L)$ must satisfy the relation

$$F(aL + b) = F(L).$$

Thus $F(L)$ is a modular type function. If we put

$$F(L) = \log\left(L - \frac{b}{1-a}\right)$$

then

$$F(aL + b) = \log a + \log\left(L - \frac{b}{1-a}\right)$$
$$= \log a + \log \beta,$$

so that

$$F(aL + b) = F(L) + \log a.$$

If we consider a periodic function of $\log \beta$, with the period $\log a$, this function will also satisfy the condition

$$F(aL + b) = F(L).$$

Therefore, the most general expression for $F(L)$ is

$$F(L) = \sum_n \left\{ A_n \sin \frac{2\pi n \log \beta}{\log a} + B_n \cos \frac{2\pi n \log \beta}{\log a} \right\}$$

This indeterminate expression is thus found in trying to define the sensation function purely on the basis of a determination of "double sensation" values.

In the particular case for which $a = 1$, we have the functional equation

$$f(L + b) = 2f(L).$$

A particular solution of this equation is

$$f(L) = 10^{mL}, \quad \text{where } m = \frac{\log 2}{b}$$

The general solution is obtained by multiplying this function by a function $F(L)$ which satisfies the condition

$$F(L + b) = F(L).$$

Hence $F(L)$ has the period b.

We now select for the sensation function the two particular solutions

$$a \neq 1: \quad S = A \times \beta^{\gamma}$$
$$a = 1: \quad S = A \times 10^{\delta}$$

where $\beta = L - \frac{b}{1-a}$, $\gamma = \log 2/\log a$, and $\delta = \frac{L}{b} \log 2$

A being a constant which is determined in such a way that when $L = 40$ decibels we have $S = 1000$. It may be noted that since

$$L = 20 \log (p/p_0),$$

(where p = sound pressure, and p_0 = threshold pressure), when $a = 1$ we obtain

$$S = A \left(\frac{p}{p_0}\right)^{6.02/b}$$

The values of a and b can be determined from the data given by Stevens and Davis[1], and indicated by the solid line in Fig. 1, which illustrates the linear relationship between L_1 and L_2 (values for which $S_1 = 2\ S_2$). We find that

$$a = 1{\cdot}10, \text{ and } b = 2{\cdot}7 \text{ [db]}$$

Therefore

$$S = A(L + 27)^{\frac{9{\cdot}371}{0{\cdot}041}} = A(L + 27)^{7{\cdot}35},$$

and since $S_{L=40} = 1000$, we have

$$S = 0{\cdot}446 \times 10^{10} \times (L + 27)^{7{\cdot}35} = (0{\cdot}039\ L + 1{\cdot}05)^{7{\cdot}35}$$

and $$\log S \simeq 7{\cdot}35 \log (L + 27) - 10{\cdot}4.$$

Fig. 2 shows as a solid line the sensation curve indicated by Stevens and Davis[2], and also includes

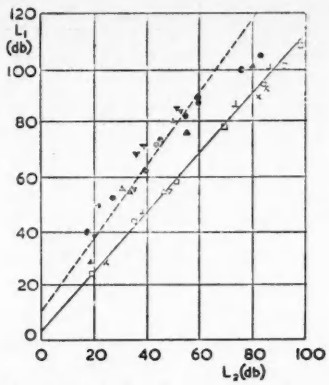


Fig. 1. **Relations between sensation levels producing acoustic sensations which are double and 10 times greater than the original sensation.**

●□ × according to Fletcher and Munson ;
▲△ „ „ Ham and Parkinson ;
▼▽ „ „ Churcher, King and Davis,

points calculated according to the formula proposed in this article, and points calculated by means of the formulae of Churcher and Knauss. Up to 100 decibels the figure shows close agreement between the calculated points and the values represented by the sensation curve.

We shall now consider the relation between two loudness levels producing the sensations $S_1 (L') = S$ and $S_2 (L) = 10\ S$. In this case

$$10\ S = A\ (L + 27)^{7 \cdot 35}, \text{ and}$$
$$S = A\ (L' + 27)^{7 \cdot 35}.$$

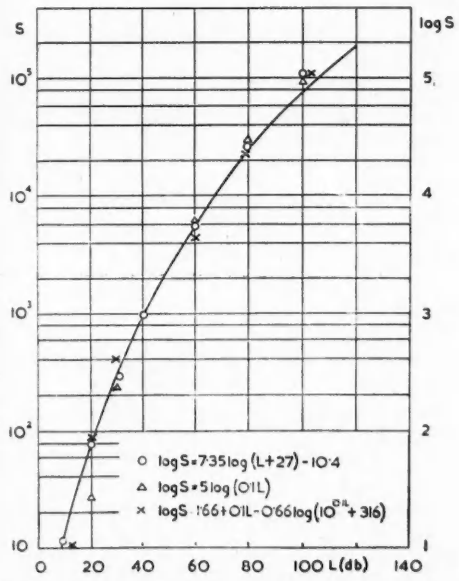


Fig. 2. **Sensation function : Solid line according to Stevens and Davis,**

Calculated points { ○○ according to proposed formula (Sacerdote), △△ „ „ Churcher, × × „ „ Knauss.

Therefore,

$$A\ (L + 27)^{7 \cdot 35} = 10\ A\ (L' + 27)^{7 \cdot 35},$$
and $$L + 27 = 10^{0 \cdot 136} \times (L' + 27)\ ;$$
hence $$L = 1 \cdot 37\ L' + 10.$$

This formula (see dotted line of Fig. 1), is substantially in agreement with the experimental results obtained by various authors.

From carefully tabulated measurements of various investigators, including Fletcher, a standard sensation curve (Fig. 2) has been established[3]. The relation between L_1 and L_2 has been obtained from this standard

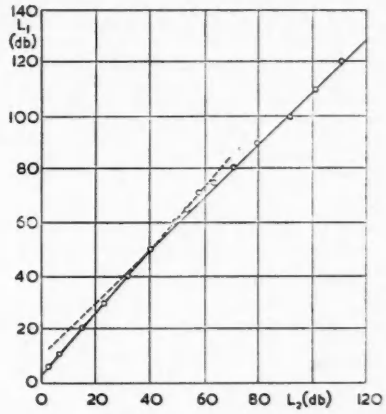


Fig. 3. **Relationship between the sensation levels producing acoustic sensations which are twice as great as the original sensation, on the basis of standard sensation tables.**

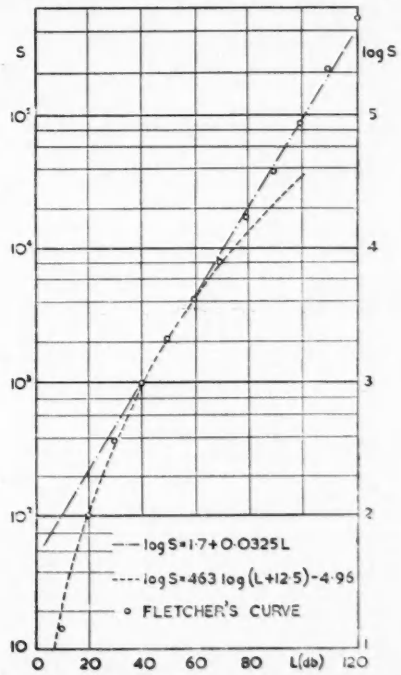


Fig. 4. **Sensation function : Standard function (according to Fletcher), and function according to the proposed formulae.**

curve, and it is found that the values of L_1 and L_2 giving $S_1 = 2S_2$ are situated on two straight line segments, as shown in Fig. 3.

For levels above 60 decibels, we have $a = 1$ and $b = 9{\cdot}3$ db, so that

$$S = 50 \times 10^{0{\cdot}0325L}, \text{ and}$$

$$\text{Log } S = 1{\cdot}7 + 0{\cdot}0325\,L\ ; \qquad \ldots \qquad \text{(A)}$$

for levels below 60 db, $a = 1{\cdot}16$ and $b = 2$, therefore

$$S = 1{\cdot}1 \times 10^{-5} + (L + 12{\cdot}5)^{4{\cdot}63}, \text{ and}$$

$$\log S = 4{\cdot}63 \log (L + 12{\cdot}5) - 4{\cdot}96. \qquad \ldots \qquad \text{(B)}$$

In Fig. 4, Fletcher's curve is indicated by circles, while the dotted and chain-dotted lines refer to the proposed equations (A) and (B). It is seen that curve (B) is valid up to 70 decibels, and curve (A) from 40 decibels upwards. The interesting point is that these curves are plotted from analytical expressions with a rational basis, avoiding purely empirical formulae.

A further question to be considered is that of the relationship between the data thus obtained and those determined from measurements of the differential threshold.

It is known that Fletcher's curve can be obtained by integrating an expression due to Weber, in which the differential threshold at a given level is proportional to the loudness level itself. Weber's law is not, however, confirmed by experiments, and even when a relation based on measured data for the differential threshold is integrated, the result is an expression for sensation which does not correspond to reality.

If we assume that the relation is

$$S = A\,(L + n)^r$$

we obtain

$$dS = (rA)\,(L + n)^{r-1} \times dL, \qquad \ldots \qquad \text{(C)}$$

an expression which gives the value of the variation of sensation due to a variation of the excitation dL.

From data published by Stevens and Davis[4], we have for the differential threshold ΔL_0 at 1000 cps :

L	5	10	20	30	40	50	60	70	80	90	100
ΔL_0	3·03	2·35	1·46	1·00	0·72	0·53	0·41	0·33	0·29	0·25	0·25

Using for the sensation function the expression already obtained :

$$S = A\,(L + 27)^{7{\cdot}35},$$

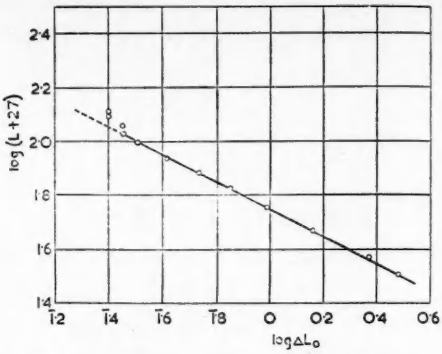


Fig. 5. **Relationship between sensation level and differential threshold.**

we find that the substitution of the values of ΔL_0 in eq. (C) does not give a constant value for dS. We do find, however, that for sensation levels up to 100 db (Fig. 5), the following law is valid :

$$\Delta L_0\,(L + 27)^2 = k,$$

where k is a constant. Denoting by ΔS_0 the variation of sensation which is hardly perceptible, and taking the previous data into account, we obtain

$$\Delta S_0 = \frac{r\,A\,k^{(r-1)/2}}{\Delta L_0^{(r-1)/2}} = \text{const. } \Delta L_0^{-3{\cdot}27} \qquad \ldots \qquad \text{(D)}$$

which signifies that the increment of sensation which is barely perceptible to the ear varies as the 3·17-th power of the differential threshold for a given sensation level.

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2. ibid. Footnote, p. 118.
3. American Standard for Noise Measurement : *Journal of the Acoustical Society of America*, 1942, Vol. 14, p. 102.
4. See Footnote 1, p. 140, of reference [2] in this article.

GERMANY

Hot-Worked Cast Iron

By E. Piwowarsky and A. Wittmoser. (From *VDI-Zeitschrift*, Vol. 91, No. 8, April 15, 1949, pp. 183-185, 7 illustrations).

The possibility of plastic working of cast iron at high temperatures, and the mechanical properties of the resulting high-quality material have been extensively investigated during the last 15 years at the Foundry Institute of Aachen Technical University. It had previously been shown that the notch effect of graphite inclusions was the main cause of the low tensile strength of cast iron, and that this strength could be considerably improved by suitable heat treatment and the development of a special cast iron with spheroidized graphite inclusions. For the production of hot-worked cast iron, slabs of cast iron are rolled, pressed, or extruded to the required profiles, at working temperatures of 1050 to 750° C. The graphite inclusions are thus orientated in the direction of working and the tensile strength along the fibres is increased very considerably. In addition, the plastic working destroys the coarse grain structure "as cast," and refines the grain by recrystallisation. The worked cast iron shows a fine sorbitic-pearlitic steel-like grain, the strength of which is further increased by the high content of silicon and manganese in the cast iron.

Static Mechanical Properties.

Fig. 1 shows the improvements in *tensile strengths* of cast iron obtained during the last few decades. The hot-rolled cast iron is clearly superior and, with its tensile strength of 55 to 75 tsi, reaches values usual for high-quality steel. The *yield stress* is about 38 to 45 tsi, i.e., 60 per cent of the U.T.S. Whilst the *elongation* of

normal and even of improved high-quality cast iron does not exceed 0·5 per cent, hot-worked cast iron shows an elongation of 2 to 5 per cent, the stronger materials also showing the higher elongation values. However, no necking could be observed in tensile tests. The *Brinell hardness* is 250 to 320, slightly higher than that of normal cast iron. The increase is due to the refined sorbitic matrix of the material, and the hardness further

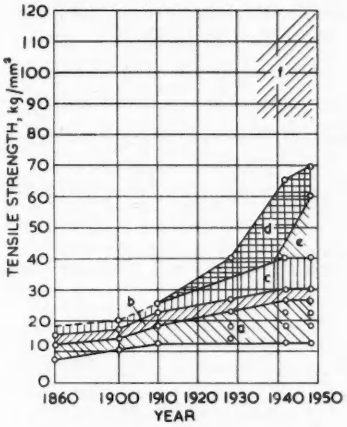

Fig. 1. **Development of the tensile strength of cast iron since the year 1860.**

a = Normal grey cast iron used for machine parts
b = high-quality grey cast iron
c = grey cast iron of special quality
d = heat treated grey cast iron
e = cast iron with spheroidized graphite inclusions
f = hot rolled cast iron.

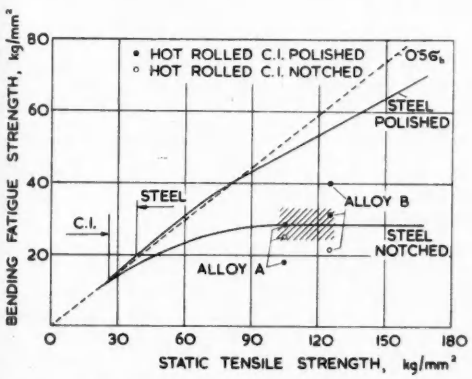

Fig. 2. **Bending fatigue strength, as a function of the static tensile strength of steel, grey cast iron, and hot-worked cast iron.**

(10 kg/mm² = 6·35 tsi).

Values obtained with hot-worked cast iron are indicated by the shaded area.

increases with increasing content of cementite. *Young's Modulus* for tension and bending is about 23 × 10^6 to 27 × 10^6 psi and slightly decreases with increasing stress. The *compressive strength*, 110 to 130 tsi, is about double the tensile strength. A *bending* test on a bar of 1·18 inches dia. and 20 diameters length showed fracture at 70 tsi stress with a final deflection of 1·2 in, whilst normal cast iron (16·5 tsi U.T.S.) fractures at 32 tsi with a deflection of only 0·5 in. The *impact strength* is also increased by hot-rolling cast iron, and was found to be 233 ft lb per sq in. for an unnotched specimen 10 × 10 × 55 mm.

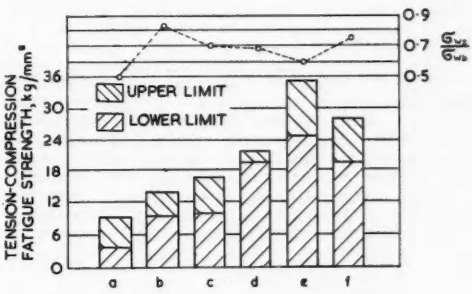

Fig. 3. **Tension-compression fatigue strength of ferrous metals.**

a = High-quality cast iron, *b* = malleable iron, *c* = annealed cast steel, *d* = 33-ton carbon steel, *e* = normalized spring steel, *f* = hot-worked cast iron.

Fatigue Properties.

The fatigue values of hot-rolled cast iron are still more favourable and are equal to those of high-quality alloy steels. The *fatigue strength in bending* is 16 to 25 psi for polished specimens. Comparative tests with steel of equal tensile strength showed that if the steel specimens are slightly notched (3 notches 0·008 in. deep, 60 deg. Vee) the same bending fatigue values are obtained (Fig. 2). This proves that the highest possible theoretical values for bending fatigue of hot-rolled cast iron are nearly reached and that further improvements can only be gained by a reduction of the graphite content, i.e., by a material more closely resembling the steel structure. Similarly notched hot-worked cast iron specimens showed only slightly reduced fatigue strength, proof of the relatively slight notch sensitivity of the material. The *tension-compression fatigue* strength is 15 to 18 tsi, nearly equal to that of normalized alloy steels (Fig. 3). The *repeated-impact* strength of cast iron, i.e., its ability to

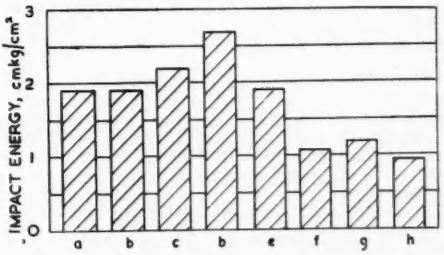

Fig. 4. **Repeated-impact strength of different ferrous metals.**

a = Grey cast iron with 0·27 per cent P
b = do. with 0·81 per cent P
c = hot-worked cast iron with 2·1 per cent C and 0·1 per cent P
d = do. with 2·6 per cent C and 0·1 per cent P
e = do. with 2·8 per cent C and 0·4 per cent P
f = cast steel with 0·35 per cent C, as cast
g = do., annealed.
h = deep-drawing sheet steel with 0·05 per cent C.

(1 cmkg/cm² = 0·47 ft-lb per sq in).

withstand an unlimited number of small impacting blows, is generally very good. Hot-worked cast iron is slightly better than ordinary cast iron and about 2·5 times stronger in this respect than cast steel or sheet steel of deep-drawing quality, (Fig. 4). The *damping capacity* of hot-worked cast iron has not been measured, but can be assumed to lie between the values for cast iron and steel.

Other Properties and Applications.

The *corrosion* properties should be about equal to those of ordinary cast iron, since the metallic structure is basically the same. The *grain structure* shows elongated graphite inclusions in a sorbitic-pearlitic matrix and, in some alloys, free cementite of rounded shape. The content of free cementite as well as the grain structure of the matrix can be varied at will by suitable alloying, heat treatment, and plastic working Piston rings made from hot-worked cast iron show the same *wear resistance* as those made from special wear-resisting cast iron alloys.

Hot-worked cast iron should, therefore, find universal application as a high-quality material. It is specially suited for machine parts subjected to wear and for structural parts required to damp down mechanical vibrations. It is a cheap alternative to the use of expensive alloy steels. Moreover, the hot-working process can be adapted for the production of special profiles. Angle profiles of 2 inches width have been successfully rolled, and promising results have been obtained with other hot-working processes, such as extrusion and forging. Special profiles can thus be cheaply produced.

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FRANCE

Influence of Stress Concentration, Speed of Deformation, and Temperature on the Rupturing Strength of Steels.

By A. GUEUSSIER and R. CASTRO.

From *Revue de Métallurgie*, Vol. 46, No. 8, August, 1949, pp. 517-536, 22 illustrations.)

I. CALIBRATION OF THE TRI-AXIAL STATE OF STRESS

Elastic Stresses. The state of stress at any point inside a material under load is completely defined.

(a) by magnitude and direction of the three mutually perpendicular *principal stresses* p_1, p_2, p_3 at that point, or

(b) by magnitude and direction of the maximum absolute value p_1 of the principal stresses and the two ratios of the state of stress, $\alpha = p_2/p_1$ and $\beta = p_3/p_1$, or

(c) by magnitude and direction of the maximum principal stress p_1 and a known symmetrical and uniform function $T = f(\alpha, \beta)$ of the two stress ratios.

The state of strain at any point of the material is completely defined by the three *principal strains*, the directions of which coincide with those of the principal stresses.

If a uniformly cylindrical test piece is loaded axially, the axial stress is $p_1 = P/A$, where P = applied load and A = cross-section. The other, transverse, principal stresses p_2 and p_3 are zero, since the axial extension results in unhindered transverse radial contractions which are calculated from the known formulae of elasticity :

$$e_2 = \frac{1}{E}\left[p_2 - \sigma(p_1 + p_3)\right] = \frac{\sigma}{E}p_1 = e_3,$$

where E = Young's modulus, σ = Poisson's ratio. If the test specimen has a sharply reduced cross-section over part of its length, the axial stress varies suddenly from the reduced to the full cross-section. Due to the continuity of the material, the transverse contractions cannot follow this rapid change and transverse stresses are set up near the transition lines, AA' and CC', Fig. 1b. Triaxial stresses are, therefore, usually investigated on notched or grooved specimens stressed in bending or tension. Assuming that at every point of the reduced cross-section the transverse strain is proportional, not to the theoretical axial stress $p_1 = -P/a$ in that section ($= a$), but to the axial stress in the full section multiplied by a correction factor C which accounts for groove effects and depends on width of groove, root radius, etc., we obtain

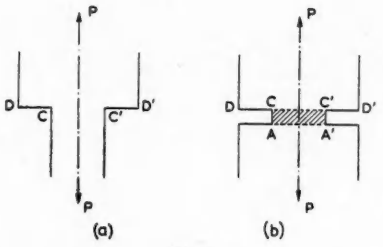

Fig. 1.

$$e_2 = \frac{C\sigma}{E} \times \frac{P}{A} = \frac{1}{E}\left[(p_2 - \sigma(p_1 + p_3)\right].$$

With $p_2 = p_3$, because of radial symmetry, and $p_1 = -P/a$, we arrive at the *fundamental equation* for the mean transverse stress in the reduced cross-section of a grooved cylindrical tensile specimen :

$$p_2 = p_3 = \frac{\sigma}{1-\sigma}\,p_1\left(1 - C\frac{a}{A}\right).$$

The stress ratios p_2/p_1 and p_3/p_1 increase with decreasing ratio a/A, i.e. increasing groove depth, and with decreasing contraction factor C, i.e. decreasing groove angle and root radius.

It has been assumed up to now that the state of stress is the same at every point of the reduced section. Photo-elastic experiments show, however, (a) that the stresses in the grooved section vary as shown in Fig. 2, and (b) that the stresses also vary from one section to the next one in the entire neighbourhood of the groove. Any alteration in any of the characteristics of the

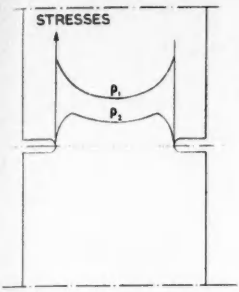

Fig. 2

groove (depth, angle, radius), therefore, widely changes the general stress pattern not only at the reduced section, but throughout the test specimen. In particular, the volume of material affected by the highest stresses is changed and the deformations as well as the energy of deformation are no longer comparable.

Speed of Deformation. In bending tests, the maximum deflection f and the maximum fibre stress p_{max} at the centre of a test bar on two simple supports and under the action of a central load are

$$f = \frac{M l^2}{12\,EI} \quad \text{and} \quad p_{max} = \frac{M}{I} \times \frac{h}{2},$$

where M = central bending moment, l = distance between supports, I = second moment of area of the test bar which is assumed to be symmetrical about the neutral axis of bending, h = maximum total height of bar cross-section. The maximum strain of the extreme fibre of the bar is

$$e_{max} = \frac{p_{max}}{E} = \frac{Mh}{2\,E\,I} = \frac{6h}{l^2} f.$$

Hence, the speed of straining is

$$\frac{de}{at} = \frac{6h}{l^2} V_0,$$

where V_0 is the speed of the impacting test load. The speed of deformation, therefore, decreases if h is reduced, i.e. if the depth of a notch at the centre is increased.

Similarly, the speed of deformation of a tensile test piece can be calculated from

$$\frac{de}{dt} = \frac{dl}{l_0} \times \frac{1}{dt} = \frac{V_0}{l_0},$$

where V_0 is again the speed of the impacting load and l_0 the test length of the specimen or the length of the reduced cross-section of a grooved specimen.

Methodical Changes in the State of Stress. It is thus shown that any change in the notch or groove shape leads to changes not only in the state of stress but also in the energy of deformation and the initial speed of deformation. If the state of stress is to be the only variable, the notch shape must remain unchanged. Consider a test piece of rectangular cross-section and given length, with a notch of given shape at the centre of its length. Assume that bending or tension forces acting symmetrically to the cross-section produce, at every point of the section reduced by the notch, a longitudinal principal stress p_1, a transverse principal stress p_2 in the direction of the depth, and a transverse principal stress p_3 in the direction of the width of section and notch. If the width is very small, contraction in this direction will be almost unhindered and $p_3 = 0$. If the width is very large, then the contraction e_3 can be regarded as being zero over the largest part of the width and p_3 reaches its maximum value $p_{3max} = \sigma (p_1 + p_2)$. For the elastic range, the following approximations can, therefore, be stated :

(a) For a given notched profile and given width, both ratios $\alpha = p_2/p_1$ and $\beta = p_3/p_1$ remain constant at any given point ; therefore, the triaxial state of stress remains unchanged for varying loads.

(b) For a given notched profile, the ratio $\beta = p_3/p_1$ increases as the specimen width increases, whilst the ratios $\alpha = p_2/p_1$, which is mainly dependent on notch shape and depth, remains nearly constant. The triaxial stress pattern increases in value with the ratio β.

The varying width of a specimen of given constant length and notched cross-section may, therefore, serve as a convenient scale for the varying value (as defined by the ratios α and β) of the elastic triaxial stress pattern. Since, for a given specimen, the stress pattern is fairly constant over the whole elastic range, one may assume even for the plastic range that a uniform change in width of otherwise identical specimens will produce an analogous uniform change in the value of the stress pattern. A similar assumption has always been made

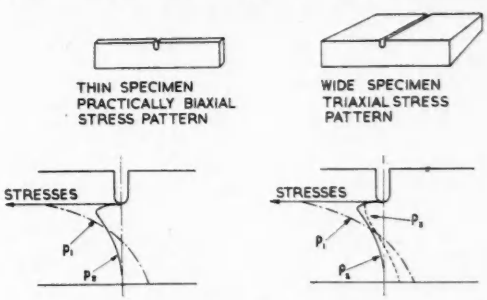

Fig. 3. **Impact test specimens. The stress curves are taken from photo-elastic experiments.**

TABLE I.

Steel	% C	% Si	% Mn	Other Elements* %	Heat Treatment
A	0·16	0·13	0·61	—	Air quench 900° C + A.Q. 875° C.
C	0·89	0·21	0·27	Cu 0·021	A.Q. 900° C + Anneal 780° C
D	0·18	0·17	0·46	Ni 2·6, Cr 0·20, Mo 0·27	A.Q. 900° C + Anneal 850° C
E	0·28	0·41	0·40	Ni 0·20, Cr 1·5, P 0·047	A.Q. 900° C + Oil Q. 875° + Temper 650° C 1 hr. in water + Sensitize 525° C, 24 hrs. in water
F	0·38	0·28	0·60	M 1·5, Cr 2·9, Mo 0·30	A.Q. 875° C + Temper 675° C + A.Q. 900° C + Temper 650° C + Sensitize 525° C, 20 hrs. in water
G	0·05	0·64	10·2	Ni 2·1, Cr 18·4, Ti 0·32	Water quench 1100° C

*S—below 0·013 %, P—below 0·024 % except where indicated.

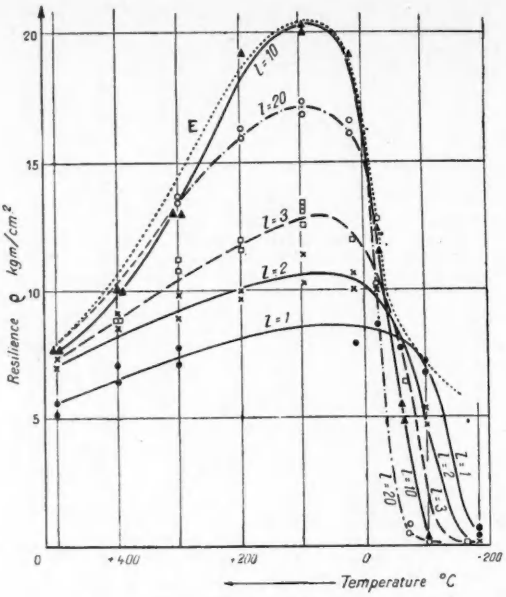

Fig. 4. **Tests on Steel A.**

in tests on specimens with varying notch shape and is even more justified in this investigation since the deformations are comparable if the width alone is changed.

II. TEST METHODS AND RESULTS.

Two Charpy impact test hammers were used simultaneously, one with a steel hammer giving a maximum impact energy of 30 kgm (217 ft-lb), the other with an aluminium hammer of 12 kgm (87 ft-lb) energy. The test supports were adjustable so that the centres of gravity of the different test pieces could be kept in the same position relative to the hammer.

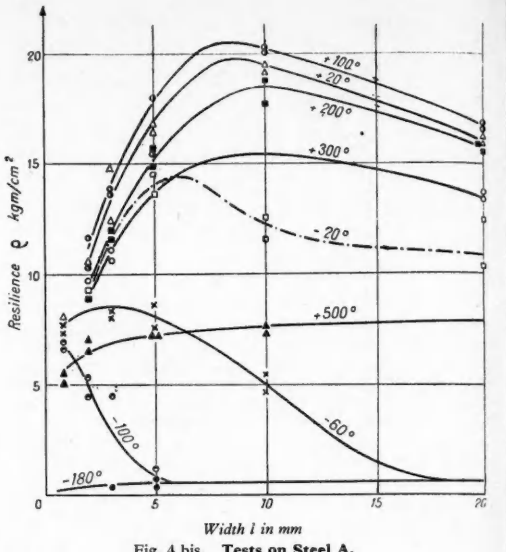

Fig. 4 bis. **Tests on Steel A.**

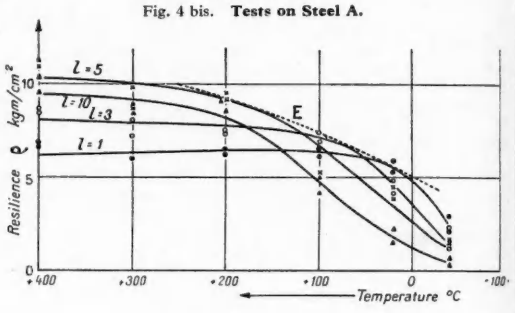

Fig. 5. **Tests on Steel C.**

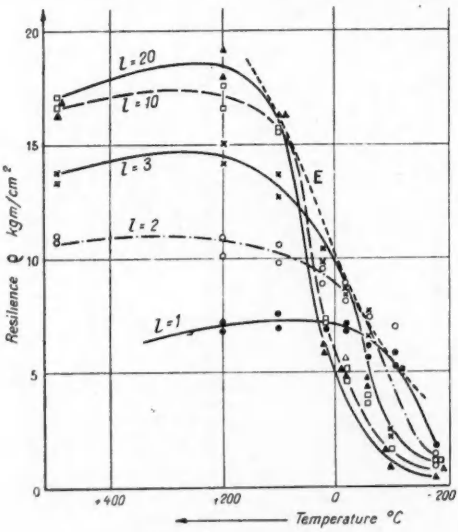

Fig. 6. **Tests on Steel F.**

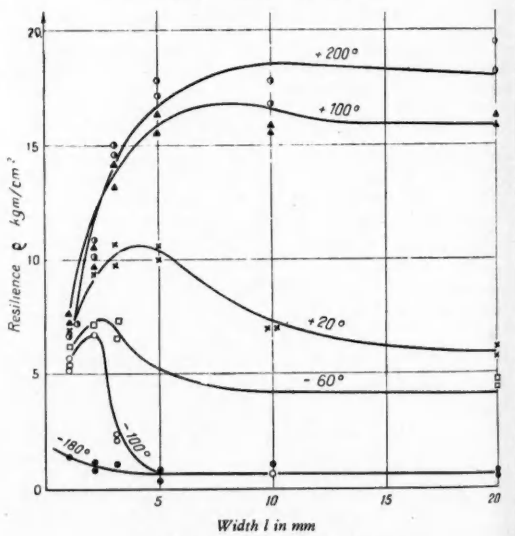

Fig. 6 bis. **Tests on Steel F.**

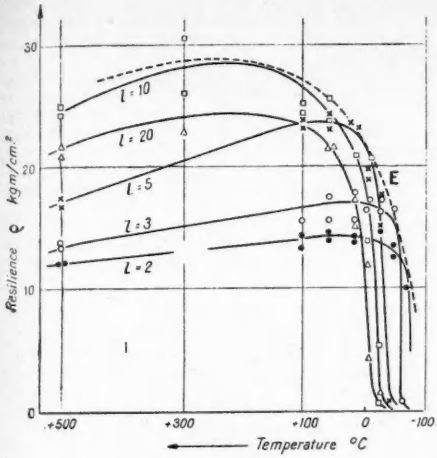

Fig. 7. **Tests on Steel G.**

Allowance was made for the different weights and energies dispersed by the throw of the broken specimens. The test materials were rolled to billets of 80 mm (3·15 in.) square cross-section, forged to 15 × 25 × 180 mm (0·6 × 1 × 7 in.) plates, heat treated to give a homogeneous structure (see Table I). The specimens (Fig. 3) were then cut out and ground to size, and the notch of 3 mm depth and 1 mm root radius milled out. Depths and lengths were kept constant and the widths varied from 1 to 20 mm. Tests were taken at various temperatures after the specimens had been held for 15 minutes in a liquid bath or gaseous atmosphere of the required temperature. The time between removal from bath and impact was never more than 3 seconds. Static tests were finally conducted at room temperature on an Amsler Bench with a knife edge of the same profile as that of the Charpy Hammers. Narrow specimens were guided in special slides in order to avoid twist. A screw transmission registered the deflection as a function of the bending load.

Impact tests were conducted at different temperatures with specimens of constant profile and growing width and hence, growing value of the triaxial stress pattern (ratios α and β). The constant profile guarantees practically identical stress gradients in the profile and identical deformation speeds (for constant impact speed) in all specimens. Even if the stress pattern changes in the course of the impact deformation, it must do so in an analogous manner in all these tests. The work done in deforming the test piece is practically identical with the rupturing energy. The rupturing energies per unit width or, since the depth is constant, per unit cross-section can, therefore, be rationally compared for all the tests of one series.

Figs. 4 to 7bis give the graphical evaluation of results* from impact tests taken at constant impact speed. If for a given width of specimen (Figs. 4 to 7 bis) the test temperature is progressively reduced from, say, + 500° C, the resilience first increases, then passes through a maximum, and finally decreases more or less suddenly in the transition zone from ductile to brittle fracture. The transition zone and the maxima are displaced towards higher temperatures as the specimen width is increased. An envelope *E* can be drawn representing the highest obtainable resiliences for a given material and impact speed at varying temperatures. If for a given temperature (Figs. 4 bis to 7 bis), the width of specimen, and, therefore, the value or

*Results from tests on Steels D and E are not reproduced here. Their graphs follow a similar pattern.—Editor, E.D.

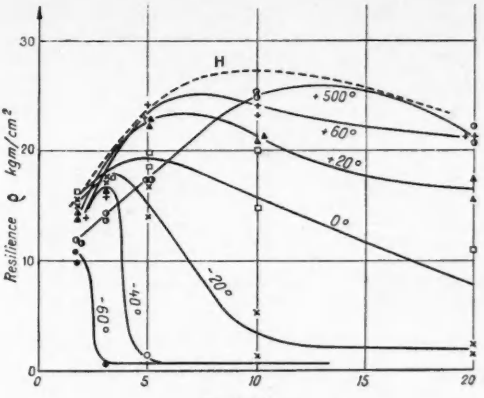

Fig. 7bis. **Tests on Steel G.**

severity of the triaxial stress pattern, is increased, the resilience first increases, then passes through a maximum, and finally decreases more or less rapidly in a zone of transition from ductile to brittle failure. This phenomenon is not always apparent in its entirety as the value of the stress pattern could only be varied over a relatively small range. The transition zone and the maxima are displaced towards larger widths, or more severe stress patterns, as the temperature increases. An envelope *H* represents the highest resilience obtainable for a given material and impact speed, for varying widths of specimen. A three-dimensional graph model may be built up showing the resilience as a function of

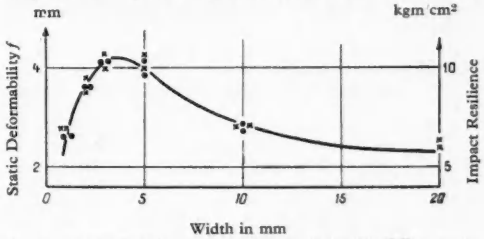

Fig. 8. **Static and impact bending tests on Steel F at room temperature.**
Crosses mark impact test results, and dots static test results. Scales are suitably chosen so that the two curves coincide.

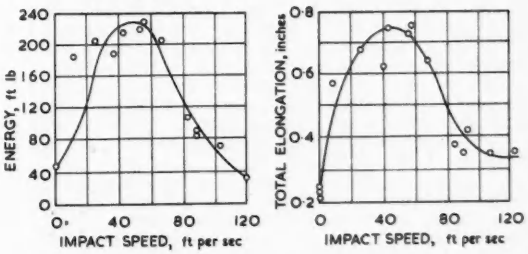

Fig. 9. **Influence of impact speed in test on drawn copper wire 0·3 in. dia., 8 in. long.**

temperature and severity of stress pattern (specimen width) for a given material and test speed. The envelopes are then the projections of the model contours onto the resilience-temperature and resilience-width planes of the graph space. The graphs show that the severity of the stress pattern (specimen width) and the reciprocal of the absolute temperature have analogous effects on the resilience.

The fractures showed the usual distinctions : fibrous fractures occur at high temperatures and low severity of stress pattern, crystalline fractures with some deformation of the specimen or without any deformation characterise the transition zone. It may be noted that some crystalline fractures with deformation may correspond to higher resilience values than fibrous fractures under different conditions.

If, in static bending tests with increasing widths of specimens, the deflection f corresponding to maximum rupturing load is taken as a measure of the deformability of the material, we obtain again a curve with a pronounced maximum, similar to the corresponding impact test curves (Fig. 8). The deformability curve is based on the fact that the constant profile produces an analogous course of deformation in all the tested specimens.

Tests with varying speeds of deformation could not be taken during this investigation. Experiments by W. H. Hoppmann on drawn copper show that the energy of deformation and the ductility pass through a maximum as the speed is increased and that the speed, therefore, plays a role analogous to that of the triaxial stress pattern and the reciprocal of the absolute temperature (Fig. 9).

(To be continued)

GERMANY

A Small Laboratory Arc Furnace

By G. Hutt and A. Schneider, Kaiser Wilhelms Institut für Metallforschung, Stuttgart. (From the Archives of the Board of Trade, German Division).

The investigation on a laboratory scale of the possibility of reducing calcium orthosilicate ($Ca_2 Si O_4$) with carbon led to the desirability of designing a furnace which satisfied the following requirements :

(1) It had to be capable of producing temperatures above 2000° C, that is to say, it had to be an arc furnace ;
(2) It had to operate either under vacuum or with an inert gas filling, as may be required ;
(3) It had to enable a simple and approximate quantitative collection of sublimates produced during the reaction ;
(4) Direct resistance heating of the melt had to be possible.

In the following, a description is given of the first design of such a furnace.

The furnace (Fig. 1) consists essentially of a water-cooled jacket of cylindrical shape, 28 cm in height, with an outer diameter of 4·5 cm. A tight-fitting carbon crucible is inserted from the bottom and is pressed in by a copper disc, which, at the same time, acts as an electrode lead. The lead itself is water-cooled, sealing being effected by a rubber joint. The second carbon electrode is inserted from the top and is adjustable. The corresponding lead is also water-cooled and sealed, and is electrically insulated from the body of the furnace by means of a rubber joint. A small inspection window close to the mounting of the electrode provides for visual observation of the chemical reaction. Inside the water-cooled jacket is a close-fitting steel tube about 18 cm long, which rests on the carbon crucible and on the inner surface of which any vapours produced during the chemical reaction condense. The melting gas is introduced slightly above the level of the rim of the crucible. The condensing cylinder is slightly indented to enable the gas to enter the furnace. The gas outlet is near the top of the jacket, it being possible to maintain a low gas pressure by means of a needle valve in the outlet.

Carbon electrodes of 6 mm and 8 mm diameter can be used, and their length can be up to 25 cm. Ignition of the arc is obtained by means of a screw head, to which the carbon holder is fixed, and by making the carbon electrode touch the carbon crucible. No burning of the carbon electrode takes place in a carefully purified and dried argon or hydrogen gas atmosphere ; therefore, it was found possible to dispense with any arrangement for adjusting the length of the electrode within wider limits.

The furnace operates on a.c. or d.c. Whereas in the case of d.c. a current intensity of 5 to 10 amperes is sufficient to produce a strong arc, intensities of between 20 and 60 amperes had to be used in the case of a.c.

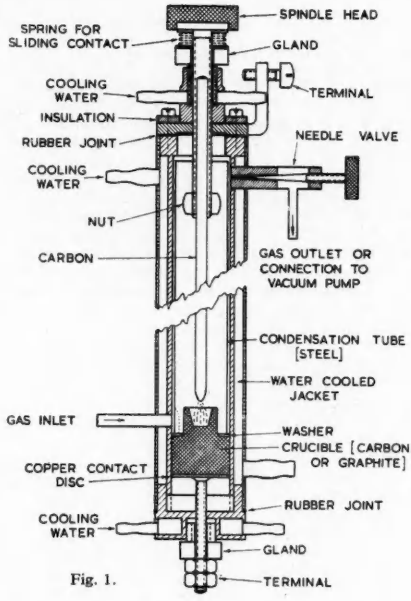

Fig. 1.

The furnace consumes 1000 to 2000 watt, and was connected to a 42-volt supply through two resistances in parallel (25 amp/1·5 ohm). In general, the thin electrodes were used for the lower intensities, and the thicker electrodes for the higher intensities.

The furnace can also be used for direct resistance heating if the electrode is kept in contact with the melt. Since, in this case, a high current intensity is required, the 6 mm carbon cannot be used. Moreover, the electrode holder in this case must be made of iron and not brass.

Ignition of the arc is brought about by brief contact between the two electrodes. In an argon or hydrogen atmosphere, the arc can easily be drawn to the required length, and due to the absence of any burning, no further adjustment of the electrode is required for a long time. However, after experiments lasting for about 3 hours, the tip of the electrode showed cracks, and chips of carbon had dropped into the crucible. On the other hand, the crucible itself had remained unchanged.

Vibrations in Power Transmission Lines—New Observations and Test Results

By M. ISBERG-HANSEN. (From *Elektroteknisk Tidsskrift*, Vol. 62, No. 15, May 25, 1949, pp. 179-186, 12 illustrations.)

1. GENERAL THEORY

FLEXURAL vibrations of transmission lines are caused by wind force on the conductors. When the wind passes over a conductor, vortices or whirls are formed on the downstream side. The whirls are shed alternatively from the upper side and the lower side of the conductor, with a frequency

$$f_w = 0{\cdot}2\ v/d \quad [\text{cps}]$$

where d = conductor diameter [metres], and v = wind velocity [m/sec]. For instance, if d = 20 mm, and $v = 3$ m/sec, the whirl frequency is $f_w = 30$ cps. While the whirls are being shed, the maximum negative pressure on the lee side shifts, with the same frequency from the upper to the lower side of the conductor, which is consequently subjected to periodic forces with, components in the plane perpendicular to the direction of the wind. These forces are generally of very small magnitude, and the force per unit length of the conductor is approximately equal to

$$F = 0{\cdot}21\ v^2\ d \quad [\text{kg/m}]$$

Thus, if d = 20 mm, and v = 3 m/sec, F = 0·378 kg/m. If the wind has the same intensity over the whole length of the line or a considerable portion thereof, the line will perform stationary vibrations in a vertical plane, with an amplitude which can be magnified by resonance effects.

The resonant frequency of a line is approximately given by

$$f_{res} = 9{\cdot}81\ P/(2qL) \quad [\text{cps}]$$

where P = conductor tension [kg], q = weight of conductor [kg/m], and L = conductor span [metres]. For instance, in the case of a steel-cored aluminium conductor of type No. 95 (Fe : Al = 1 : 6), if L = 259 m, q = 0·613 kg/m, and P = 1380 kg, then f_{res} = 0·29 cps. The nodes are situated at the ends of the line, at the points of attachment. In practice, the line rarely vibrates at its fundamental frequency but at one of its higher harmonics, $n\ f_{res}$ (n being the order

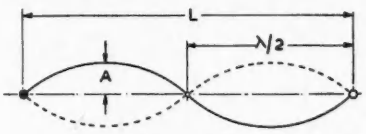

Fig. 1. **Conductor span with stationary wave.**
$\lambda/2$ = Distance between two nodes; A = amplitude; L = conductor span.

of the harmonic). In this case there will be n stationary wave crests and $n + 1$ nodes separated by a distance $\lambda/2 = L/n$ from one another. Fig. 1 shows an example with two stationary wave crests on the conductor span. The amplitude of these waves can be regarded as a harmonic function

$$A = A_0 \sin \omega t$$

where t = time [sec], A_0 = peak amplitude [cm], and $\omega = 2\pi f_w$ [rad/sec]. Furthermore, the attachments of the conductor, the insulator strings and cross-arms, are also capable of vibrating, although generally at different frequencies. These vibrations dissipate part of the excitation energy and thus have a damping effect on the primary vibration of the conductor, while also causing periodic amplitude fluctuations in some cases.

2. NEW OBSERVATIONS AND TEST RESULTS

Extensive tests involving observations over considerable periods of time were made on the seven lines of A/S Glommens Traesliberi coming from the Kykkelsrud hydro-electric station. On six lines, copper conductors of 50, 70 and 95 mm² cross-section were used, while for the seventh line on which most of the tests were carried out, steel-cored aluminium conductors of type No. 95 were installed. The conductors were stressed up to 40-50 per cent of the nominal breaking load at 25° C.

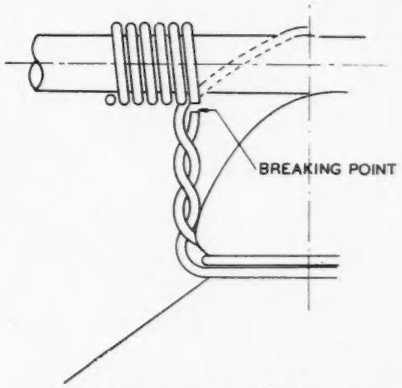

Fig. 2. **Breakage due to bending vibrations.**

Conductors without dampers or other protective devices are apt to break as a result of the vibratory stresses. Breakage generally occurs at the points of attachment (see Fig. 2). The test line with steel-cored aluminium conductors was, therefore, provided with twelve Hoffmann type vibration dampers. The line had a total length of approximately 33 km and the conductors, with a mean span of 193 metres, were mounted on towers with three cross-arms. The tests showed that the dampers were most effective when situated in the immediate vicinity of the towers. In spite of the improvement thus obtained, it was found that there was still sufficient vibration to cause appreciable metallic chatter. The following observations were made during the tests:—

1. Large amplitude vibrations occurred over 30 per cent of the total line length, low amplitude vibrations over 29 per cent, occasional vibrations over 25 per cent, and no vibrations over 16 per cent of the total line length.
2. Large amplitude vibrations appeared almost exclusively in spans over open, flat or sparsely covered terrain. Minor oscillations occurred in slightly undulated terrain. Occasional vibrations of both types were observed over wooded areas where the conductors were situated at or above the tree top level. Vibrations were negligible in forests and other sheltered districts.
3. Vibrations were seldom noticed in spans with tensioning elements at one or both ends.

Following these tests, a new type of damper was designed (see Fig. 3). Dampers of this type were mounted on spans where the vibration amplitudes were

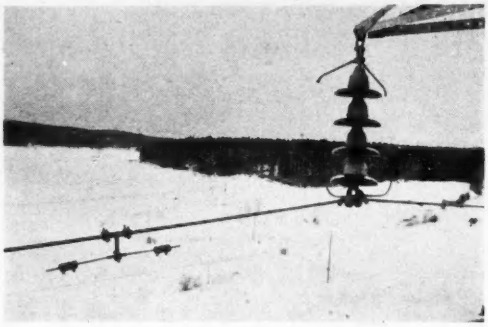

Fig. 3. **New type of vibration damper.**

greatest. At first, two dampers were used per phase, one at each end of the span, at a distance of 1·75 m from the cross-arm. Tests showed, however, that one damper per phase was sufficient to take care of all vibrations except those of unusually high amplitude. Altogether 372 dampers have to date been installed.

It was also found that, in general, all the conductors were subject to vibrations, and that conductors with high tensile loads vibrated more than slack conductors. Copper conductors were less subject to vibrations than the steel-cored aluminium conductors.

Further tests were undertaken in order to obtain more extensive information on the behaviour of the steel-aluminium conductors. The test line consisted of three conductors of type No. 95, Fe : Al = 1 : 6·15, 26 aluminium strands ; 7 steel strands total cross-section 175·7 mm², total weight 0·613 kg/m, overall diameter 17·23 mm, breaking strength 5650 kg. One of the phases was used as reference line for purposes of comparison. The reference line and one of the other lines were stressed with a preload equal to 60 per cent of the nominal breaking stress, while the third phase was mounted without preloading. All vibrations, wind velocities and wind directions were registered over considerable periods of time by means of relays and recording instruments. The test span had a length of 259 m. The conductors were mounted on sheave blocks to avoid stresses due to temperature variations, and the preload was applied at one end by means of weights. The voltage source was the 220-volt d.c. network. The recordings indicated the occurrence frequencies and amplitudes of vibration, as shown in Fig. 4, and tests were continued over a period of about one year.

An investigation was also made of the relationship between the static preload of the conductor and the vibration time. Fig. 5 shows the relative vibration time (expressed as a percentage of the total time elapsed during the test) plotted as a function of the preload of the conductor, for the lines tested. Line 1, mounted on the upper cross-arm, was subjected to a tensile load of 1380 kg and used as reference line throughout the test. Line 2, on the second cross-arm, was subjected to tensile loads varying between 640 and 1380 kg. Line 3, on the lower cross-arm, was not preloaded

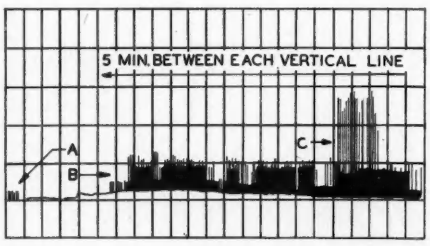

Fig. 4. **Recording of vibration amplitudes and frequency occurrence.**

A = vibration amplitudes of 1·5-2·7 mm.
B = " " " 2·8-4·9 mm.
C = " " " 5·0 mm or more.

These are half-total amplitudes. The vibrations were measured at a distance of 0·75 m from the point of attachment.

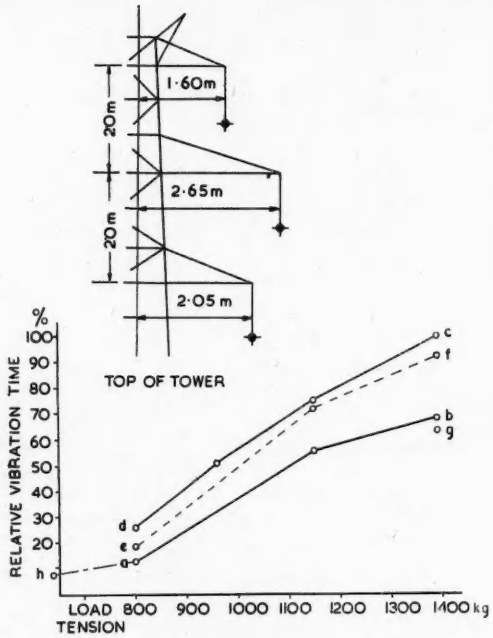

Fig. 5. **Relative vibration time as a function of the load tension.**

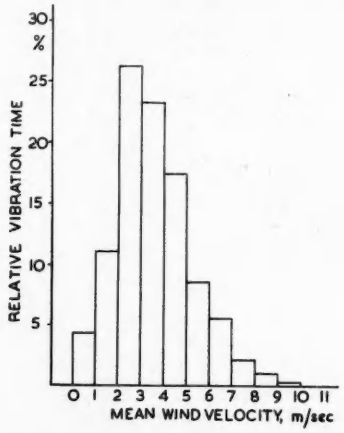

Fig. 6. **Effect of wind velocity on vibrations.**

Measurements effected on conductor No. 1, subjected to a constant tension of 1380 kg. Only vibrations with a half-total amplitude exceeding 1·5 mm were recorded.

apart from some tests in which the load was varied between 800 and 1380 kg. In a second test, line 2 was mounted on the lower cross-arm, and line 3 on the middle cross-arm. The results of these tests, given in Fig. 5, only take account of vibrations with a half-total amplitude not less than ± 1·5 mm, measured at a point situated at a distance of 0·75 m from the supporting point.

The results of the tests may be summarised as follows:—

1. There is always the possibility that vibrations will occur in steel-cored aluminium conductors, particularly if the tensile load is not much lower than 700-800 kg, even with maximum summer temperatures. The vibrations increase in duration with higher stresses. The damping due to the cross-arms can have a considerable influence on the vibrations (see curves *d–c* and *a–b*, and points *b* and *g* in Fig. 5).
2. Vibrations can appear even when the direction of the wind varies, and the vibration amplitudes tend to increase when the wind direction is more or less perpendicular to the conductor.
3. Vibrations are initiated at wind velocities between 0 and 1 m/sec, and can occur even at velocities below 0·5 m/sec. Vibration occurrence reaches a maximum when wind velocities are between 2 and 4 m/sec. The vibrations cease when wind velocities exceed 9–10 m/sec, although vibrations of short duration have occasionally been recorded at 12·4 m/sec, (see Fig. 6).
4. The vibration times are greater during the winter months, and the conductors vibrate more at night than during daytime. The vibrations are most pronounced during the hours of sunrise and sunset.

ITALY

Theoretical and Experimental Results on the Fitting of Rotors on Shafts of Electrical Machines

By F. BURLANDO. (From *L'Ingegnere*, Vol. 22, No. 5, May, 1948, pp. 390-396, 7 illustrations.)

THE object of this investigation is to develop formulae by means of which the stresses and pressures occurring in rotor hubs fitted on alternator shafts may be readily determined. Tests on various rotor types, carried out by the experimental department of the Società Nazionale Officine Savigliano, of Turin, have given results in close agreement with the theoretical values.

1. GENERAL THEORY

We consider an element of the rotor hub as shown in Fig. 1. The strains in the radial, tangential, and

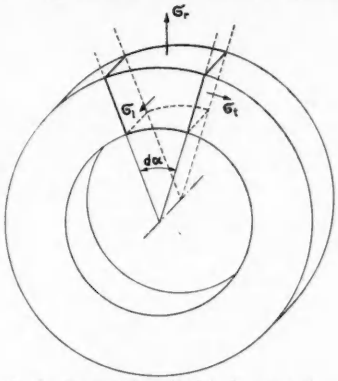

Fig. 1. **Stresses in cylindrical rotor hub.**

longitudinal directions (denoted by the subscripts r, t, and l), are given by :

$$\epsilon_r = \frac{d\Delta r}{r}, \quad \epsilon_t = \frac{\Delta r}{r}, \quad \epsilon_l = \frac{d\Delta l}{dl} \quad .. \quad (1)$$

The corresponding resultant stresses in these directions are, therefore,

$$T_R = E\,\epsilon_r = \sigma_r - \frac{1}{m}\,[\sigma_t + \sigma_l] \quad .. \quad (2a)$$

$$T_T = E\,\epsilon_t = \sigma_t - \frac{1}{m}\,[\sigma_l + \sigma_r] \quad .. \quad (2b)$$

$$T_L = E\,\epsilon_l = \sigma_l - \frac{1}{m}\,[\sigma_r + \sigma_t] \quad .. \quad (2c)$$

E being the elasticity modulus, and m Poisson's ratio. The addition of equations (2a) and (2b) gives

$$\epsilon_r + \epsilon_t = \frac{1}{mE}\,[(\sigma_r + \sigma_t)(m\text{-}1) - 2\,\sigma_l] = 2B \quad .. \quad (3)$$

Multiplying both sides by $r\,dr$, we obtain :

$$r d\Delta r + r\Delta r = 2\,Br\,dr, \text{ or } d(r\Delta r) = B\,d(r^2),$$
$$\text{hence } r\Delta r = Br^2 + C,$$

so that $$\Delta r/r = \epsilon_t = B + (C/r^2) \quad .. \quad .. \quad (4)$$

and, therefore, by virtue of equation (3)

$$\epsilon_r = d(\Delta r)/r = 2B - [B + (C/r^2)] = B - (C/r^2) \quad (5)$$

Substituting the value of ϵ_l, ϵ_t, and ϵ_r in eqs. (2), and using the abbreviation $\epsilon_l = A$, we obtain

$$\left.\begin{aligned} \sigma_r &= \mu E\,[(m\,B + A) - (m-2)\,(C/r^2)] \\ \sigma_t &= \mu E\,[(m\,B + A) + (m-2)\,(C/r^2)] \\ \sigma_l &= \mu E\,[2B + (m-1)\,A] \end{aligned}\right\} \quad (6)$$

where $$\mu = m/[(m+1)(m-2)].$$

If $K_h = R_h/r_h$ = ratio of external hub radius to the internal hub radius, p_f = fit pressure due to elastic reaction of shaft, and p_e = external pressure due to reaction of the rotor arms, so that

$$p_f = \sigma_r, \text{ and } p_e = \sigma_t,$$

then the values of the constants C and $mB + A$ can be obtained as :

$$\left.\begin{aligned} C &= \frac{1}{\mu\,(m-2)\,E}\,M_h R_m^2 (p_f - p_e) \\ mB + A &= (p_f - K_h^2\,p_e)\,M_h/(\mu E) \end{aligned}\right\} \quad (7)$$

where $M_h = 1/[K_h^2 - 1]$ and $K_h = R_h/r_h$.

The stress σ_l in the axial direction is negligible in comparison with the values of σ_r and σ_t. The substitution of eqs. (7) in eqs. (6) gives for the stresses at a radius r from the shaft centre

$$\left.\begin{aligned} \sigma_r &= -M_h[(p_f - p_e)\,K_r^2 + (K_h^2\,p_e - p_f)] \\ \sigma_t &= +M_h[(p_f + p_e)\,K_r^2 - (K_h^2\,p_e - p_f)] \end{aligned}\right\} \quad (8)$$

where $K_r = R_h/r$.

Inserting these values in eqs. (2) gives

$$\left.\begin{aligned} T_R &= \frac{2M_h}{3}[2(p_f - p_c)K_r^2 + (K_h^2 p_c - p_f)] \\ T_T &= \frac{2M_h}{3}[2(p_f - p_c)K_r^2 - (K_h^2 p_c - p_f)] \end{aligned}\right\} \quad (9)$$

The maximum values at the inner hub surface of the resultant stresses are obtained for $r = r_i$:

$$T_{R\,max} = \frac{2M_h}{3}[(2K_h^2 - 1)p_f - K_h^2 p_c] \quad .. \quad (10a)$$

$$T_{T\,max} = \frac{2M_h}{3}[(2K_h^2 + 1)p_f - 3K_h^2 p_c] \quad .. \quad (10b)$$

The minimum values, obtained when $r = R_0$, are

$$T_{R\,min} = \frac{2M_h}{3}[(p_f + (K_h^2 - 2)p_c] \quad .. \quad .. \quad (11a)$$

$$T_{T\,min} = \frac{2M_h}{3}[3p_f - (K_h^2 + 2)p_c] \quad .. \quad .. \quad (11b)$$

The tangential stresses T_T are tensile stresses which increase the overall diameter, whereas the radial compressive stresses T_R decrease the cross-sectional area. When $K_h^2 p_c - p_f > 0$, optimum utilisation of the material is achieved under compression if the compressive load at the elastic limit ρ is

$$\rho = T_{R\,max} \quad .. \quad .. \quad (12)$$

Substituting eq. (12) in eq. (10a) gives the corresponding fit pressure

$$p_f = \frac{3}{2}\rho[1 - \tau] + \tau p_c \quad .. \quad (13)$$

where $\tau = K_h^2/(2K_h^2 - 1)$.

When $K_h^2 p_c - p_f < 0$, optimum utilisation of the material is obtained if the tensile load at the elastic limit is

$$\theta = T_{T\,max} \quad .. \quad .. \quad .. \quad (14)$$

Eq. (10b) then gives

$$p_f = [\theta\mu/(2M_h^2 K_h^2)] + \mu p_c \quad .. \quad (15)$$

where $\mu = 3K_h^2/(2K_h^2 + 1)$.

For steel hubs, we always have $\rho = \theta$. Furthermore, generally $T_T > T_R$, since the reaction p_c of the rotor arms and disc is always much lower than the fit pressure p_f of the shaft. Unless the shaft is hollow, which is seldom the case, the formula (15) can be applied to most cases. For hollow shafts, however, the stress conditions are calculable by means of eq. (13).

2. CALCULATION OF THE INTERFERENCE FIT

The interference fit will now be calculated, first of all for the case of a hollow shaft. Let

R_s, r_s = external and internal radii of shaft,

$K_s = R_s/r_s$ = shaft radius ratio.

During the fitting process, the internal hub surface is subjected to a tangential strain ϵ_{th}, and the external shaft surface to a radial strain ϵ_{rs}. The interference fit between the shaft and hub is, therefore,

$$i = 2[\epsilon_{th} r_h - \epsilon_{rs} R_s] \quad .. \quad .. \quad .. \quad (16)$$

hence, $i = 2[r_h(T_{T\,max}/E_h) - R_s(T_{R\,min}/E_s)]$ .. (17)

Substituting the values of $T_{T\,max}$ and $T_{R\,min}$ from eqs. (10b) and (11a), we have

$$i = 2\left[\frac{2r_h M_h}{3E_h}(N_h p_f - 3K_h^2 p_c) - \frac{2R_s M_s}{3E_s} Q_s p_f\right] \quad (18)$$

where $M_{h,s} = 1/[K_{h^2,s} - 1]$, $N_h = 2K_h^2 + 1$, and $Q_s = K_s^2 - 2$.

With sufficient approximation, we can write $D = 2R_s = 2r_h$, and, therefore, the relative interference for hollow shaft is

$$\frac{i}{D} = \frac{2}{3}\left[\frac{M_h}{E_h}(N_h p_f - 3K_h^2 p_c) + p_f \frac{M_s}{E_s} Q_s\right] \quad (19)$$

For a solid shaft, $r_s = 0$, so that in this case

$$\frac{i}{D} = \frac{2M_h}{3E_h}(N_h p_f - 3K_h^2 p_c) - \frac{1}{3E_s} p_f \quad .. \quad (20)$$

Conversely, for a given value of i/D, the fit pressure for a hollow shaft is

$$p_{f\,hollow} = \frac{3E_s}{2M_s\left[\frac{E_s M_h}{E_h M_s} N_h + Q_s\right] \times \left[\frac{2K_h^2 M_h}{E_h^2} p_c + \frac{i}{D}\right]} \quad .. \quad (21a)$$

while for solid shafts

$$p_f = \frac{3}{2} \times \frac{1}{\left[\frac{M_h N_h}{E_h} - \frac{1}{E_s}\right]} \times \left[\frac{M_h K_h^2}{E_h} p_c + \frac{i}{D}\right] \quad (21b)$$

3. EFFECT OF THE ROTOR ARMS

The elastic reaction p_c of the rotor arms on the hub depends on the size and shape of the rotor. For disc type rotors, we have $p_c \simeq 0$. When the hub is very thick compared with the shaft diameter ($K_h > 2{\cdot}0$ — $2{\cdot}5$), the rotor arms do not take part in the deformation work during the fitting of the shaft, so that here also we can put $p_c \simeq 0$.

Thus, the calculation of p_c is only necessary in the case of rotor arms on a relatively thin hub, the assumption then being that the outer circumference of the rotor is not deformable owing to its much thicker section and to the impossibility of the stresses reaching up to it along the rotor arms. The value of p_c is calculated by equating the deformation of the arms due to compression to the tensile deformation of the outer diameter of the hub.

If the rotor arms are interconnected, their deformation as a result of the fitting process is given by

$$\epsilon_r = p_c S_h L/(S_r E_r) \quad .. \quad .. \quad (22)$$

where L = length of a rotor arm, S_r = cross-sectional area of an arm in a plane perpendicular to L, and S_h = area of the outer surface of the hub.

If the arms are not interconnected, the deformation is less than previously, and is due to the deformation work occurring only in the zone of contact of the arms and the hub. In this case we have

$$\epsilon_r' = p_c L/E_r \quad .. \quad .. \quad (23)$$

The work which is not absorbed by the hub appears in the form of shear stresses on the remaining free surface of the hub. For computations, it is thus preferable to use in this case eq. (22).

The radial deformation of the external surface of the hub of radius R_h is

$$\epsilon_r = T_{R\,min}/E_h \quad .. \quad .. \quad (24)$$

(*Concluded on page* 371)

WORKSHOP TECHNOLOGY

The Negative Spring—A Basic New Elastic Member

By W. J. Cook and P. C. Clarke. (From *Product Engineering, U.S.A.*, Vol. 20, No. 7, July, 1949, pp. 136-140.)

A NORMAL extension spring is characterised by a force-deflection curve with a positive gradient, i.e., increased force must be applied to obtain increased deflection. The new patented spring member named the "Neg'ator" and produced by the Hunter Spring Co. Inc., of Lansdale, Pa., U.S.A., has a flat or even negative characteristic (see Fig. 1).

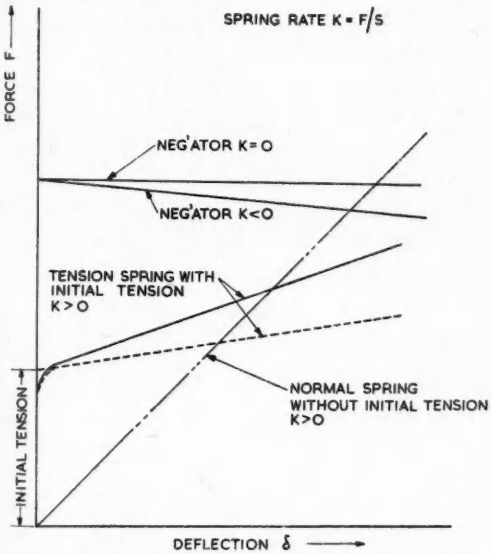

Fig. 1. **Force-deflection characteristics of Neg'ator springs and normal compression and tension springs.**

Owing to its prestressed condition, the spring does not deflect until the force has reached a certain critical value. When this value is reached, however, the Neg'ator spring deflects suddenly over a considerable distance, without applying any additional force (flat characteristic). The spring can also be made in such a way that after obtaining initial deflection by reaching the critical load value, further deflection will occur even when the load is reduced (negative or receding characteristic).

The Neg'ator is capable of acting efficiently around corners and through small openings. It has usable deflections 30-50 times the original dimensions of the unloaded spring. Furthermore, it can store and deliver approximately twice as much energy as an ordinary spiral or power spring occupying the same space.

The spring is designed as a tight coil of flat material which is highly prestressed during manufacture, so that each spring element of the tape possesses a strong natural curvature (see Fig. 2). The spring force can be calculated by means of the following ap-

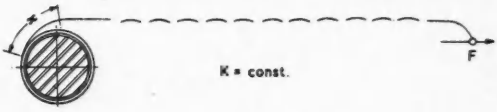

Fig. 2. **Neg'ator spring shown as a series of spring elements, each having its own natural radius of curvature.** The section marked "x" is the resisting length of the spring in this position.

proximate formula, obtained by equating the external work done in extending a spring element to the energy stored in the spring by straightening, i.e., by increasing the radius of curvature:—

$$P = \frac{EI}{2}\left[\frac{1}{R_n^2} - \left(\frac{1}{R_n} - \frac{1}{R_1}\right)^2\right]$$

where P = applied force [lb], E = elasticity modulus [psi], $I = bt^3/12$ [in^4] = moment of inertia of spring cross-section (b being the spring width, and t the spring thickness [in]), R_n = natural radius of curvature of the spring element [in], and R_1 = radius of mounting (spool, shaft, etc.) [in].

The spring can be operated in two different ways: (1) by applying a pulling force at its free end, the wound portion of the coil being free to rotate with the shaft or drum on which it is mounted, or (2) by having the spring wound on spools at both ends, and applying an uncoiling torque in one direction by forcibly rotating one of the two spools.

In both cases, the resisting torque which opposes the uncoiling pull is developed only by the portion of the spring which is being straightened (see Fig. 2, where the resisting length of the spring is denoted by "x"). Contrary to what is the case with normal springs, the other portions of the coil (the section

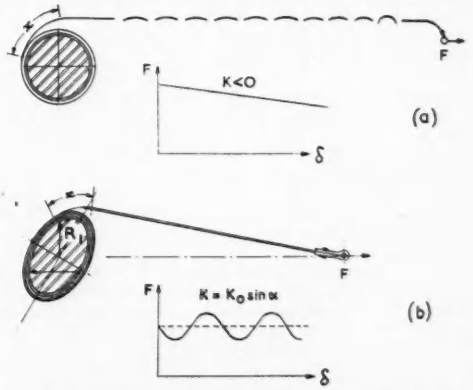

Fig. 3a and 3b. **Spring with negative characteristic, and spring with sinusoidal characteristic.**

already straightened and the remainder of the spring still wound on the original spool) do not contribute to the resisting torque. This explains the flat characteristic of this spring, since the resisting length of the spring is substantially constant on a spool of given diameter. A negative characteristic is obtained by giving a smaller natural radius of curvature to the section of the coil nearer the free end than to the material in the inner section of the coil (Fig. 3a).

Further unusual features of this spring are illustrated by the following properties. If the Neg'ator spring is wound on two spools of different diameters, it will run to the smaller spool unless it is held by an external force, since the spring exerts less force on the large diameter spool. If the spring is wound on an elliptical spool (Fig. 3b), the spool curvature varies during its rotation, and the resisting torque will also vary cyclically, giving a sinusoidal characteristic.

The Neg'ator spring can be used not only as a normal tension or compression spring, but also as an end grip, an automatic coiling or clamping device, a friction band, and generally as an energy or motion transmitting element, either alone or in conjunction with other normal spring systems. It can also be applied as a helical tape for binding or wrapping, as a force-compensating device for vernier adjustment, and as a means of converting linear into circular motion. The possibility of using it in electromagnetic circuits, for instance for changing the amount of laminated core iron in a magnetic field, is also being considered.

Rapid-Reversing Drive for Planing Machines

By M. STEINEBRUNNER. (From *Brown Boveri Review, Switzerland*, Vol. 36, Nos. 3/4, March/April, 1949, pp. 126-133, 6 illustrations.)

THE Brown Boveri Company, of Baden, Switzerland, has developed a new rapid-reversing drive for planing machines of all sizes, and other industrial machines involving rapidly reciprocating masses. The drive is designed to meet the following requirements : infinitely variable speed control over a wide range, stable operating speeds, high table speeds and cutting power, rapid, smooth and accurate reversals of the table, simple control, and long life.

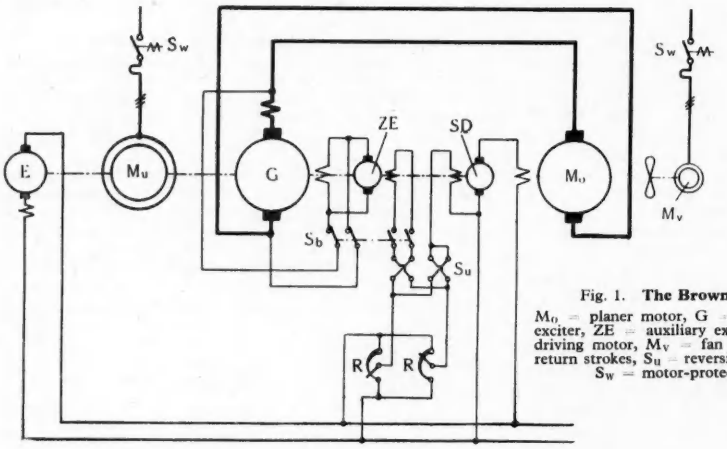

Fig. 1. **The Brown-Boveri rapid-reversing drive.**
M_o = planer motor, G = variable-voltage generator, E = main exciter, ZE = auxiliary exciter, SD = booster, M_u = generator driving motor, M_v = fan motor, R = rheostats for forward and return strokes, S_u = reversing contactors, S_b = braking contactor, S_w = motor-protecting switch for 3-phase motors.

This drive has been evolved from the Ward-Leonard controlled drive which has a d.c. variable-speed reversing motor connected to separate control and power supplies. The motor armature is permanently connected to a variable-voltage generator, while the field winding is fed by an exciter, which also supplies, either directly or via a second auxiliary exciter, the excitation for the variable-voltage generator. The diagram of Fig. 1 gives details of the electrical system, which incorporates the principles of the Ward-Leonard control.

The speed of a d.c. motor can be varied by varying either (1) the field intensity, or (2) the armature voltage. Field variation gives speed control over a range of 1 to 4, whereas armature voltage variation permits speed control from maximum speed practically down to zero. Although the potential remaining across the unexcited generator does not always permit regulation down to zero rpm, the motor can be brought to rest by a brief reversal of generator excitation which suppresses the residual field until the armature voltage has completely vanished. To reverse the excitation, the field winding of the generator can be used as a suicide winding, reversed in polarity and shunted across the armature, or alternatively, a separate brake winding can be provided.

These two methods of d.c. motor speed control give different torque and output characteristics, as shown in Fig. 2. With armature voltage regulation, the torque is constant over the entire speed range, so that motor output rises linearly with speed, provided the generator driving motor is powerful enough. When field regulation is used, the motor output remains constant, while the torque drops off with a hyperbolic characteristic as speed increases.

For convenience, regulation of the armature voltage

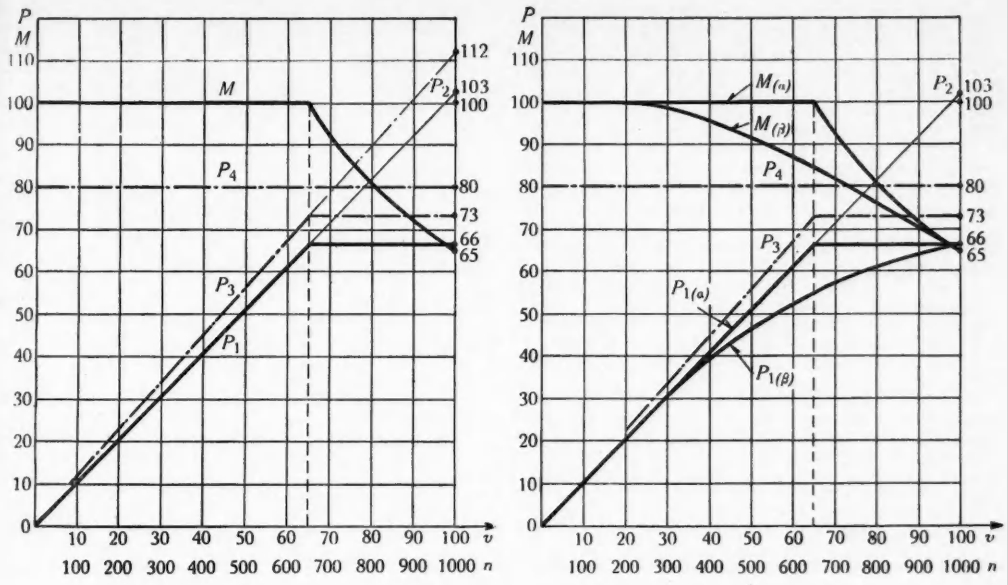

Fig. 2. **Characteristics of planer drive with " straight " and " mixed " Ward-Leonard control.**

P_1 = power applied to table by planer motor (kW), as a function of the table and motor speeds.
P_2 = rating of planer motor.
P_3 = output of variable-voltage generator as a function of its voltage.
P_4 = rating of generator driving motor.
M = torque of planer motor,
n = rpm of planer motor.
v = speed of planer table (cutting speed).

(a) Straight Ward-Leonard Control

Max. table speed : v = 100 m/min
Max. power applied to table (= rating) : $P_{1\,max} = P_2$ = 103 kW.
Max. output of variable-voltage generator : $P_{3\,max}$ = 112 kW.
Rating of generator driving motor : P_4 = 80 kW (approx.).
Max. torque of planer motor (rated torque) : M = 100 kgm up to n = 650 rpm (or v = 65 m/min).
Minimum torque of planer motor : M = 65 kgm at n = 1000 rpm (or v = 100 m/min).

Note.—In this example the cutting power is limited by the rating of the generator driving motor, which accounts for the torque dropping off in the upper speed range.

(b) Mixed Ward-Leonard Control

Rating : P_2 = 103 kW.
(α) *Successive armature and field control :*
Max. power applied to table : P_1 = 66 kW with v = 65-100 m/min.
Output of variable-voltage generator : P_3 = 73 kW.
(β) *Simultaneous armature and field control :*
Power applied to table : P_1 = 56 kW with v = 65 m/min. or P_1 = 66 kW with v = 100 m/min.
Torque of planer motor : M = 100 kgm for $n \leq 300$ rpm or $v \leq 30$ m/min. M = 85 kgm for n = 650 rpm or v = 65 m/min. M = 65 kgm for n = 1000 rpm or v = 100 m/min.

In all cases, the planer motor must be designed for the rating P_2.

alone will be termed " straight W.L. control," regulation of both armature and field voltages " mixed W.L. control," and armature regulation at low speeds followed by field regulation at higher speeds " successive control." The " mixed control " system is the simplest and most practical method, and gives torque and output characteristics similar to those obtained with " successive control " (see Figs. 1 and 2).

The reversal of the direction of rotation of the d.c. motor is obtained either directly, by reversing the polarity of the variable-voltage generator, or for rapid-reversing drives by means of an auxiliary exciter. With mixed speed control, the field of the motor must also be strengthened at the right moment. The control power required is extremely small for the size of the drive used. The auxiliary exciter is used for reversing control and speed stabilisation.

The Ward-Leonard controlled d.c. motor is reversed with negligible losses, as, apart from small frictional and current losses, the kinetic energy of the moving parts is recuperated during the deceleration process. It is thus particularly suitable when reversals take place in rapid succession with heavy workpieces and high table speeds, resulting in large amounts of kinetic energy being stored up in the parts in motion.

CHOICE OF RAPID-REVERSING DRIVES

The choice of a rapid-reversing drive for planing-machine tables is governed by the following factors :

(1) the maximum force required for cutting and to overcome friction, i.e., the maximum force to be applied to the table ;
(2) the cutting speed corresponding to the maximum driving force and the maximum table speed ;
(3) the efficiency of the gearing between the table rack and the planer motor ;
(4) the inertia of the table, the workpiece, and the gearing.

Items (1) to (3) are important for speed control, while (4) must be known in order to adapt the drive to the reversing process.

The rating of the planer motor depends mainly on the specified maximum driving force on the table and the maximum table speed, even when this force is only applied in the lower speed range. Since motor reversal is more difficult and less rapid at higher running speeds, there will be a definite maximum speed limit, which should not be exceeded to maintain optimum conditions.

It is always advisable to adopt liberal dimensions

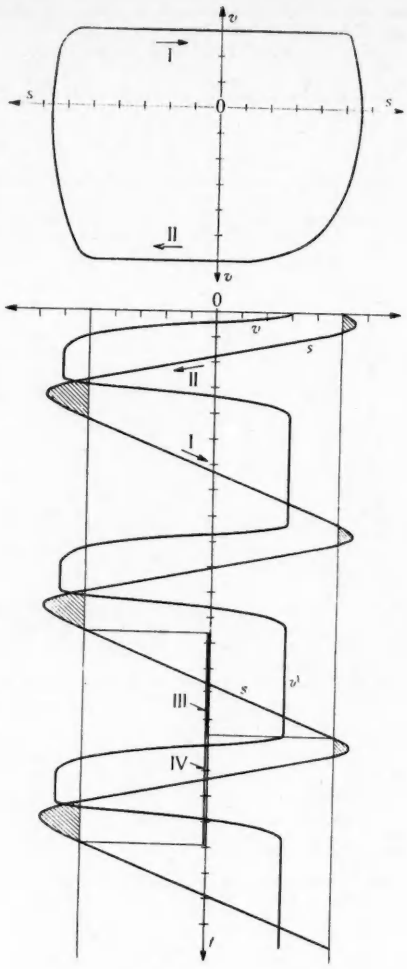

Fig. 4. Relation between table travel and speed for a single work cycle (upper diagram). **and during a number of successive work cycles** (lower diagram).

By suitable adjustment of the cams initiating the reversal of motion at each end of the stroke, the time required for a work cycle can be reduced to a minimum, with sufficient time still remaining for the auxiliary motions (feed and tool lifting). Symbols : I = cutting stroke, II = return stroke, III = working or cutting time, IV = idling time. Scales : t-axis : 1 division = 0·5 sec ; v-axis : 1 division = 0·166 m/sec ; s-axis : 1 division = 0·1 metre.

for the variable-voltage generator and the generator driving motor, since both with high-speed tools and with the carbide tools which are being increasingly employed on planing machines, it is an advantage to take heavy cuts.

Fig. 2 shows the relationship between table driving power and motor rating at maximum speed, and also the planer motor torque plotted against motor speed, for straight and mixed Ward-Leonard control systems. It also shows the outputs of the variable-voltage generator and the generator driving motor.

Most planing machines cut only in one direction, and the reciprocating motion of the table then consists of alternating cutting and idle return strokes. The speed of the return stroke is adjusted to the maximum permissible value in order to reduce the idling time, and during the return stroke only frictional forces (which are a function of the frictional coefficient and the weight of the table and work) have to be overcome.

With machines cutting in both directions, the return stroke also becomes a cutting stroke, and the production of the machine is correspondingly increased. The mean load on the table drive is then also higher. For machines cutting in one direction, the mean load on the table drive, referred to the total working time, is 60-70 per cent of the maximum cutting power, whereas it may reach 90-100 per cent for machines cutting in both directions.

The full working cycle of the table consists of the forward and return strokes, and the two reversal processes at the ends of the strokes. The reversing process must be as rapid as is compatible with the mechanical characteristics ensuring adequate reliability. A measure of the reversing time (i.e., the time required to decelerate or accelerate the table, work, gearing, and

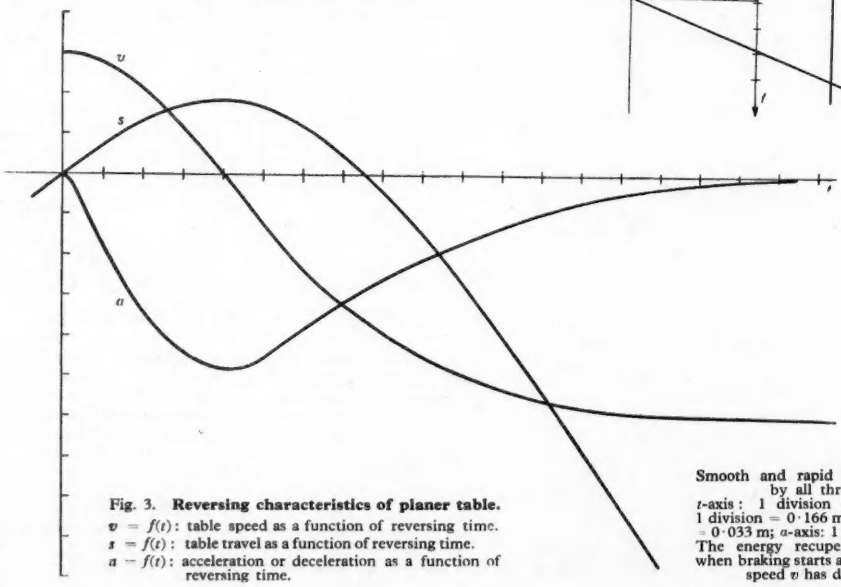

Fig. 3. Reversing characteristics of planer table.

$v = f(t)$: table speed as a function of reversing time.
$s = f(t)$: table travel as a function of reversing time.
$a = f(t)$: acceleration or deceleration as a function of reversing time.

Smooth and rapid reversal is evidenced by all three curves.
t-axis : 1 division = 0·05 sec ; v-axis : 1 division = 0·166 m/sec ; s-axis : 1 division = 0·033 m; a-axis: 1 division = 0·83 m/sec². The energy recuperation process begins when braking starts and ends when the table speed v has dropped to zero.

the table motor) is obtained from the relationship between the total inertia of the system and the motor torque.

Assuming that the accelerating and decelerating torques of the motor are constant, and neglecting friction effects, the deceleration time (in seconds) is given by the equation

$$t = 2{\cdot}67 \times 10^{-3} \times n \times WD^2_{\text{total}} \,/\, T \quad \ldots \quad (1)$$

where WD^2_{total} = total swinging moment = $g \times$ total moment of inertia of the moving parts [kg m²], n = motor speed [rpm], T = decelerating or accelerating torque [kg m].

The total WD^2 of the table, work, gearing, and coupling, relative to the motor shaft axis, is

$$WD^2_{\text{total}} = \frac{1}{\pi^2} W_{\text{TW}} \frac{v^2}{n^2} + \sum \left(WD_{\text{G}}{}^2 \frac{n_{\text{G}}{}^2}{n^2} \right) + WD_{\text{PM}}{}^2 \quad (2)$$

where W_{TW} = weight of table and work [kg], v = table speed [m/min], n = speed of planer motor [rpm], $WD_{\text{G}}{}^2$ = swinging moment of gearing [kg m²], n_{G} = speeds of gearing [rpm], $WD_{\text{PM}}{}^2$ = swinging moment of the planer motor [kg m²].

Eq. (1) shows that the reversing time is shorter for low values of WD^2_{total}, low motor speeds, and high starting and decelerating torques.

Short reversing times can be obtained by using low-speed planer motors with a moment of inertia only a fraction of that of standard motors having the same rating. In small planing machines with a maximum table speed of 60-70 m/min, the motor inertia still represents about 2/3 of the total moment of inertia, and the table and the work together barely attain 1/3. The ratio of the inertia of the table and work to the total inertia increases, however, if the table speed is raised to 100-120 m/min, and the total moment of inertia is, therefore, also greater. Longer reversing times are unavoidable with shorter cutting and return times, unless the motor reversing torque is increased and the table and gearing inertias are reduced by using lighter designs without impairing their mechanical strength.

The character of the reversing process is clearly shown by oscillograph records. Fig. 3, which is based on such records, shows the travel, speed, and acceleration of the table, plotted as functions of time, and the curves obtained are smooth curves, without any sudden jumps. The deceleration of the table is gradual, and has its maximum value at a low table speed. Thus slight gear backlash can be taken up without shock at the beginning of the reversing movement.

In electrical machines it is also an advantage, from the point of view of stresses, if the reversing torque and current reach maximum values when the motor speed and voltage are low. As shown in Fig. 3, the energy recuperation period begins when deceleration commences, and ends when the table speed has dropped to zero. The reversing torque developed by the motor is largely expended in the rotor of the motor itself, and the remainder, further reduced by the torques required to overcome the gear inertias, is applied to the table. This gradual reduction of the reversing torque due to the inertias of the various parts of the drive increases the efficiency of transmission during the reversing process, as compared with conventional transmissions. However, it also results in a low reversing torque on the table which hardly reaches the torque corresponding to the full driving force, in spite of the rapid and powerful reversing motion.

Fig. 4 shows the variation of table travel and table speed with time during several successive work cycles. A speed-travel diagram for a single cycle is also given. These graphs enable the relation between working and idling times to be determined. From the time-travel diagram, it is apparent that the highest efficiency of the cycle is attained when the reversing impulse is given at each end of the stroke, in such a manner that the times for the auxiliary motions (feed and tool lifting) are reduced to the minimum permissible values.

CONSTRUCTION OF ELECTRICAL EQUIPMENT

The machines constituting the Ward-Leonard drive are provided with overload protection, and are adjusted to give full reliability and accuracy over a period of many years. In the event of a failure of a.c. supply, the planer motor is instantaneously braked, since the converter absorbs and stores up the energy released during deceleration. All heavy-duty components, such as contactors, automatic reversing and stop switches, etc., are designed to withstand at least 10 million operations. The control switches, instruments, auxiliary motors, and the a.c. apparatus for the generator driving motor, can be combined in a control desk. A pendent pushbutton station makes it possible for the operator to control the electrical equipment from different positions.

Assembling Metal Parts by Shrink Fitting

By C. DESCHARS. (From *Materials & Methods, U.S.A.*, Vol. 29, No. 5, May, 1949, pp. 64-67, 4 illustrations.)

THE assembling of metal parts by fitting can be effected by three simple methods :—

1. *Pressure fitting :* The parts are fitted together by means of pressure supplied by a press of sufficient capacity. In this process, the contacting surfaces are affected (milling phenomena), but their structure and composition are not altered. Particular caution is required when using materials with tendencies of brittleness.
2. *Expansion fitting :* This is achieved by heating the female part to increase its diameter before inserting the male part. The necessary grip is provided by the female part, after cooling ; heat distortion should be avoided.
3. *Contraction or shrink fitting :* By sufficient cooling, the dimensions of the male part contract until it can penetrate into its location. The part warms up subsequently to room temperature and exerts the necessary contact pressure by expanding to its original dimensions.

These methods can be combined or used simultaneously for particular applications.

For shrink fitting, the male part is immersed in a cooling fluid, and for this purpose liquid nitrogen is frequently used. Liquid nitrogen is obtained industrially from liquid air, from which the nitrogen is removed in fractionative columns. Liquid nitrogen has a low boiling point of —320·3° F, and, unlike oxygen, it does not ignite in the presence of greased parts. The liquid is shipped in special spherical containers with double walls and an intermediate vacuum. To avoid building up a high internal pressure by evaporation,

the immersion container is never sealed but provided only with a screen lid which prevents the introduction of foreign matter and reduces the rate of evaporation. The container has a long specially designed neck for the immersion of large parts.

Liquid nitrogen has the same density and colour as water, and boils with heavy white fumes produced by the instantaneous condensation of atmospheric impurities. When a metal is immersed in the liquid, the boiling is first relatively slow until the gaseous layer surrounding the metal is dissipated. The metal then becomes sufficiently cold to be wetted by the liquid, and increased boiling occurs. This ceases when the metal reaches the temperature of the liquid.

Provided the operator does not touch metallic parts cooled by immersion, occasional contact of the skin with liquid nitrogen is generally harmless, as the skin is protected by the thin layer of air surrounding it. In practice, the parts to be fitted are immersed without direct manual contact, by means of a special jig, and the operator is provided with gloves.

The quantity of liquid nitrogen required to cool the immersed part varies according to the heat given up by the metal or alloy used. The consumption of liquid nitrogen in shrink fitting various metals is given in the following table.

TABLE I.

Material	Density lb./in³	Average specific heat between + 60 and — 320° F.	Litres of liquid nitrogen required to cool 1 lb. of metal
Steel	0·285	0·110	0·249
Aluminium	0·094	0·175	0·396
Bronze	0·323	0·081	0·183
Copper	0·322	0·079	0·179
Tin	0·264	0·086	0·194
Iron	0·284	0·085	0·192
Cast iron	0·260	0·120	0·271
Brass	0·305	0·081	0·183
Magnesium	0·063	0·208	0·471
Manganese	0·280	0·093	0·210
Nickel	0·317	0·087	0·197
Silicon	0·080	0·118	0·267
Zinc	0·258	0·084	0·190

The density and specific heat of a given alloy can be calculated using the mixture law when accurate figures are needed.

The linear contraction coefficients of metallic parts cooled by liquid nitrogen are given in Table II. The cooled metal should easily slip into the outer part. The contraction determined from Table II must allow the fit to take place. Tolerances for sliding fit are given in Table III.

APPLICATIONS OF SHRINK FITTING

Sleeves for Car Engine Cylinders : With the press fitting process previously used, there was considerable wastage, as the nitrided cast-iron sleeves were rather brittle and only 0·8 in. thick. Expansion heating with the engine block heated to 570° F was an expensive process.

Shrink fitting by immersion was found simpler, quicker and more economical. The jig required for the operation was made of brass, which is less brittle than other materials at low temperatures. The sleeve was fixed between the two trays of the jig, which was provided with a wooden holder. After the fitting, the jig was removed by unscrewing the bottom tray. The sleeve was 2·5 in. in diameter, 0·8 in. thick, and 3·8 in. high, and the required interference fit was 0·0016 in. The fitting required 1 litre of liquid nitrogen for a 6-cylinder engine, and the operation took 8 min. per engine, or 50 per cent less than by former methods. Micrographic examination of the fitted sleeves showed no apparent changes in structure.

TABLE II.

Material	Linear coefficient of expansion (between + 60 and — 320° F.)	Linear contraction of 1 inch of length
Bronze	$8·9 \times 10^{-6}$	$3·4 \times 10^{-3}$
Copper	7·9 „	3·0 „
Tin	12·1 „	4·6 „
Iron	5·0 „	1·9 „
Cast iron	4·7 „	1·8 „
Brass	8·9 „	3·4 „
Magnesium	11·8 „	4·5 „
Manganese	8·7 „	3·3 „
Nickel	5·5 „	2·1 „
Silicon	0·95 „	0·36 „
Zinc	14·5 „	5·5 „
High-speed steel* A	5·0 „	1·9 „
„ „ „ B	5·3 „	2·0 „
„ „ „ C	4·5 „	1·7 „
„ „ „ D	5·8 „	2·2 „
Tool steel E	5·0 „	1·9 „
„ „ F	5·3 „	2·0 „
„ „ G	4·7 „	1·8 „
„ „ H	5·3 „	2·0 „
Cr. V. steel SAE6150	5·8 „	2·2 „
Machine steel SAE1020	5·8 „	2·2 „
Stainless steel 18 Cr–8 Ni	7·4 „	2·8 „
Aluminium 2S	8·2 „	3·1 „
„ 24–ST	9·3 „	4·0 „
Invar 36	1·3 „	0·5 „
Cast alloy I	3·7 „	1·4 „
„ J	4·5 „	1·7 „
Carboloy Grade 44A	1·8 „	0·7 „
Monel metal	6·6 „	2·5 „

High-speed steels :* (A**) 18 % Ni, 4 Cr, 1 Va ; (**B**) 6 Ni, 5 Mo, 4 Cr, 2 Va ; (**C**) 18 Ni, 4 Cr, 2 Va, 9 Co ; (**D**) 5 Ni, 4 Mo, 4 Cr, 4 Va ; *Tool steels :* (**E**) 1·1% C ; (**F**) 0·9 C, 1·2 Mg, 0·5 Cr, 0·5 Ni ; (**G**) 0·5 C, 0·9 Cr, 1·25 Ni ; (**H**) 2·25 C, 12 Cr, 1 Mo ; *Cast alloys :* (**I**) 20 Co, 8 Ni, 7 Mo, 5 Cr, 2 Va, 0·7 C, 0·7 B, balance Fe ; (**J**) 44 Co, 17 Ni, 33 Cr, 2·25 C, 2 Fe.

TABLE III.

Diameter of the fitting in	Total tolerance of sliding fit, in.	
	close	moderate
0·5	0·0005	0·0009
1·0	0·0008	0·0017
3·5	0·0016	0·0035
6·0	0·0025	0·0050

Sleeves for Cylinders on Hydraulic Presses : Sleeves with an outer diameter of 18·5 in., an inner diameter of 17·7 in., 19·7 in. high, and weighing 128 lb, were shrink-fitted in the four cylinders of hydraulic presses. Expansion fitting would have required intense heating of the block and produced deformations due to the release of foundry stresses. The sleeves were suspended by means of lifting rings, and stopped in the right position by a shoulder at the bottom. The four sleeves were fitted in 25 min., and 150 litres of liquid nitrogen were used.

Seating of Valves for Car Engines : Shrink fitting allows good centring and a solid grip of 0·004-0·006 in., and is used by many French motor-car manufacturers. The tendency is to use either smooth or threaded valve seats of chromium-molybdenum cast iron with cast-iron cylinder heads, and bronze seats with aluminium heads. The jig consists of a brass rod with a wooden handle and a brass base, with a slotted elastic ring underneath, around which the seat is forced. The brass rod extends under the base to allow of easy centring. The 120 exhaust valves in 30 blocks can be laid in 30 min., using approximately 3 litres of liquid nitrogen.

A special machine for fitting valve seat inserts has been designed by the Ford Motor Co. The machine is located in the assembly line. The engine block is automatically fixed on each side by guided rollers. Eight ducts coming from the refrigerator arrive in

(*Concluded on page 368*)

PRODUCT DESIGN AND DEVELOPMENT

Redesign of existing "Dough Sheeting and Crossrolling" Machine from a Cast Iron Construction to a Welded Steel Construction

By J. BOEHLER. (From the book "*Design for Welding*," published by the James F. Lincoln Arc Welding Foundation, 1948, Chapter XXIV, pp. 885-890, 5 illustrations).

THIS machine is used to roll and crossroll the dough for soda crackers before being baked. The reasons for redesigning this machine from cast iron to a welded steel construction are several: (1) To obtain a modern, streamlined machine; (2) to allow of fully enclosing all moving parts, thereby increasing safety factors; (3) to strengthen the structure of the machine. These objectives could not be accomplished in a very practical manner with cast iron frames.

Since this machine is built, according to requirements, for either left or right-hand discharge of dough, and also for a variety of conveyor belt widths ranging

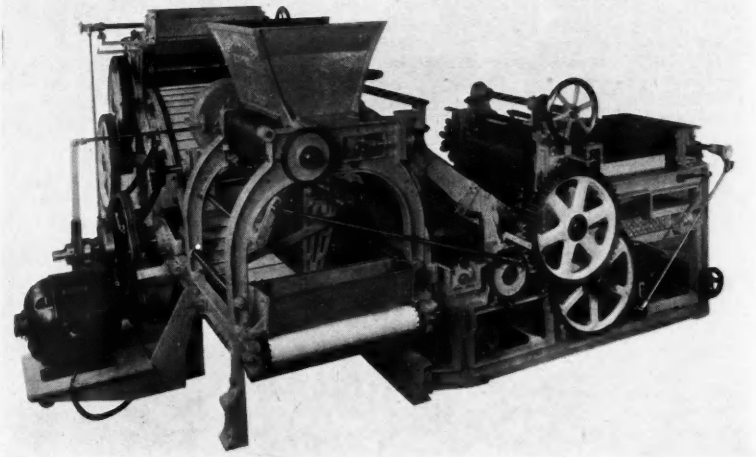

Fig. 1.
Front view of R. H. Dough-sheeting and Crossrolling Machine with cast iron frame before re-designing.

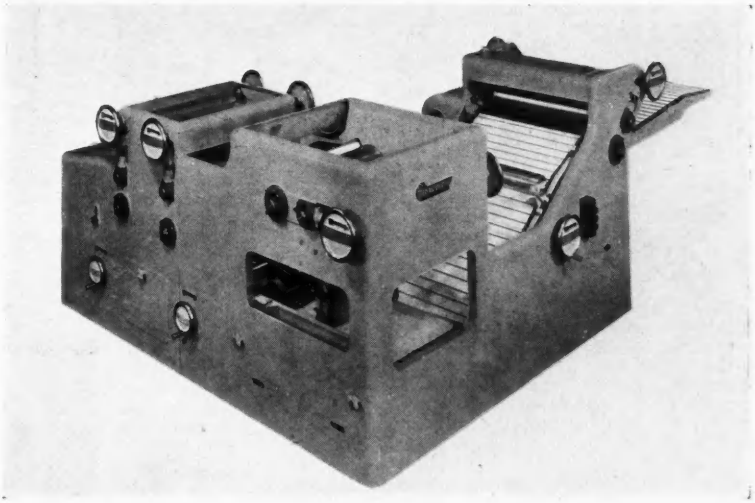

Fig. 2.
Front view of L. H. Dough-sheeting and Crossrolling Machine with welded steel construction after re-designing.

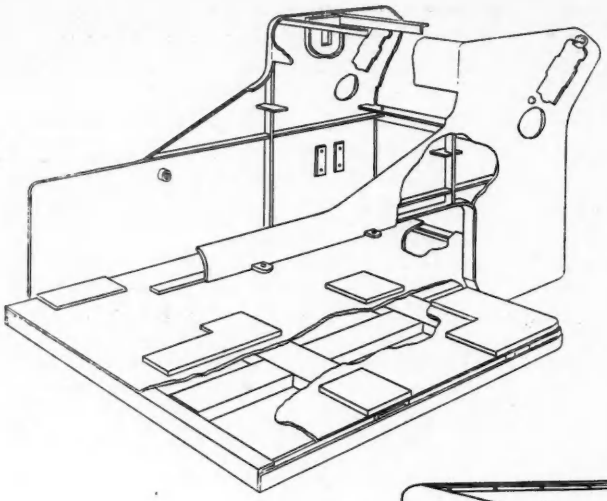

Fig. 3. **Perspective view of frame unit No. 1.**

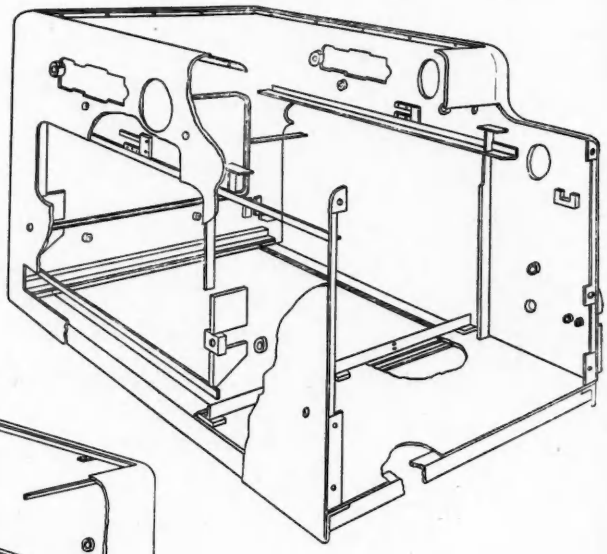

Fig. 4. **Perspective view of frame unit No. 2.**

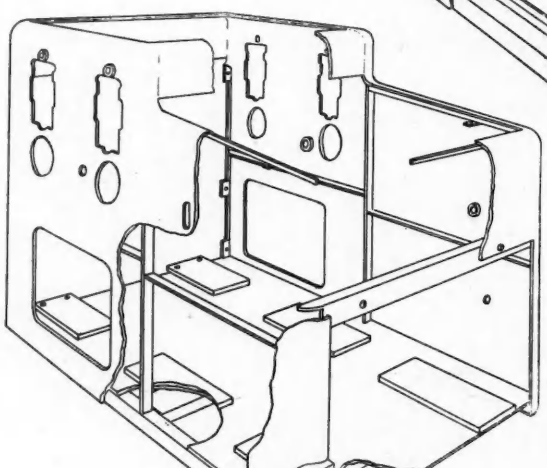

Fig. 5. **Perspective view of frame unit No. 3.**

from 24 in to 40 in., the pattern cost involved for each individual job was quite appreciable. In conversion to a welded design, this cost was greatly reduced. The original cost of wood patterns for a machine of this type has been estimated to be about $2,600. All the cast iron frames, struts, and bases would have to be machined and all holes drilled and reamed, following which the unit would have to be assembled. The cost of a welded steel frame, including all struts and base plates, is $1,150.

In using a welded frame construction, no machining of frame parts, struts, and bases is required; only a series of small holes of up to $\frac{3}{8}$ in. is hand-drilled. A few bosses having $\frac{3}{4}$ in. and 1 in. holes are individually made up with undersize holes, welded in place with the aid of a jig, and then line-reamed by hand, thus eliminating costly machining.

For a general picture of the process of redesigning this machine, reference should be made to Fig. 1. This photograph shows a front view of the dough sheeting and cross-rolling machine of cast iron frame construction before redesigning.

Fig. 2 shows a front view of the same machine (this illustration, however, shows a left-hand machine, while Fig. 1 is a right-hand machine), after it had been redesigned for welded steel construction, with all moving parts fully enclosed.

To facilitate fabrication of the frame, it is made in three sections or units. Figs. 3 and 4 are perspective views of units Nos. 1 and 2 respectively. After units 1 and 2 are fabricated, they are welded together as one single unit before the drivers, rollers, conveyors, etc., are installed within. Fig. 5 is a perspective view of frame unit No. 3. In this unit the drives, rollers, conveyors, etc., are also installed; the unit is then attached by bolts to units 1 and 2.

The main reason for bolting this unit to the others is to facilitate shipping, which, since the machine is of L-shaped construction, might otherwise be difficult.

The overall dimensions of the completely assembled machine are 9 ft 2 in. × 10 ft 6 in. × 5 ft 5 in. high and the total weight is 10,000 lbs. From the illustrations the general construction of this machine can be seen. The base parts are made of heavy angle and channel iron, and are covered with $\frac{3}{8}$ in. boiler plates with $\frac{1}{4}$ in. thick pads welded thereon to form a strong foundation for the heavy "Reeves Motodrives" installed in the lower section. The uprights or frames are fabricated from $\frac{3}{8}$ in. thick boiler plates, with a rim of 3 in. radius on all outside edges to provide round corners; and wherever extra strength is required, ribs, pads, and bosses are welded in place. All cross-struts are welded in place, thus making the frame one solid box construction. Many functional parts, such as flour boxes, conveyor take-ups, mounting brackets, etc., are also welded directly into the framework. Several other removable brackets and conveyor frames are of welded steel construction or cut out from steel plates.

In addition, the heavy rollers previously made from cast iron are now fabricated from steel tubing with steel hubs welded to the ends. The heavy roller bearings for these rollers are of the flange type, the bearings being inserted into oversize holes and secured tight after alignment.

Since this machine has not yet been put into large-scale production it has so far been impossible to establish a more exact cost than that indicated by the limited data given in this paper.

Design for Sales

A MOST IMPORTANT BUT OFTEN NEGLECTED ASPECT OF INDUSTRIAL PRODUCT DESIGN

By K. D. Burbidge.

Enough has been said of the dollar crisis, of hard and soft currency areas, of tariffs and restrictive practices and of tottering standards of living to make the British public fully aware that they have reached a crossroads in their national progress. On the solution of these problems depends Britain's standing as a free nation in the world. Yet it sometimes appears that her people have allowed themselves to be lulled into an almost passive sense of security by the tremendous loans lavished on them. The danger is now that they will miss the simple truths and the straightforward solutions that lie behind this facade. However this may be, there can be no denying that a sound approach to the solution of this crisis, that will not compromise Britain's independence unduly, must necessarily include the balancing of her import-export ledgers. She must export services and goods to the outside world to at least equal the value of her essential imports, and she must do this in what is fast becoming an absolute buyers' market. This is no insoluble problem; nevertheless, effective solution will require highly efficient and practical design, production and marketing techniques from our industry.

The purpose of these notes is to single out the product design side of industry for a re-examination of its scope and content as a means of demonstrating a void that must be filled if the design picture is to be completed. Failure to note this void must surely prejudice the sale of British products abroad and allow many potential wins to go by default, particularly in the U.S., the stronghold of our principal competitors.

While no business-man will deny that product design is a necessary element of industrial processes, and while indeed provision is made for it in all industrial concerns that undertake fabrication work, nevertheless, when attempts are made to define the limits of the word and to check on the processes necessarily involved in its practice, the wide differences in its uses and the limited extent of the provision actually made for it become apparent. If one asks a production engineer what he means by "design," he will almost certainly talk first of the steps taken by the design side to speed production and simplify processes by special attention to design detail. If one asks a mechanical engineer he will discourse on matters relating to mechanical effort, efficiency and control. A salesman will talk of the customer's long term satisfaction with such things as "performance quality," "handling quality," "wearing quality" and "styling" and the points that make for sales in a sharply competitive market. These are each capable of infinite expansion if full justice is to be done to the tremendous technical knowledge that lies behind each single aspect; yet, stated simply, they form the basis for a comprehensive outline of the scope of "industrial product design" (see Table overleaf).

This breakdown into aspects of production, mechanical and sales design is important. Unless a clear distinction is maintained between these three throughout the design stages, adequate attention will not be given to all of them. The tradition of a sellers' market and immediate practicability too often result in attention being concentrated mainly on "straight engineering." Modern industrial techniques inevitably require specialisation if attention is to be adequate. Specialisation involves the employment of suitably qualified designers.

Production Design.	*Mechanical Design.*	*Design for Sales.*
Factors governing Quantity of Materials and Man hours.	Factors governing efficiency of mechanical functions.	Factors governing functional requirements.
Factors governing Quality of Materials and Workmanship.	Factors governing efficiency of mechanical operation.	Factors governing operational requirements.
	Factors governing "styling" (mechanical).	Factors governing "styling" (visual).

Note.—In this outline the word "function" has been taken as being defined as "essential use" and is considered from two points of view: the requirements of the market (as analysed). and the efficiency of the mechanical means adopted to supply these requirements. The word "styling" has been taken as including both "visual styling" and "mechanical styling," which are complementary.

The day of the jack-of-all-trades is long past and the engineering sides and the sales side must each be given the continuous attention they require. This in spite of the fact that the immediate reaction produced by increased competition is to cut costs by every means possible. Nevertheless, when attempts are made to plan for special concentration on the "functional" and "operational" requirements' aspects of product design, it soon becomes evident that these are very largely controlled by engineering considerations. Their efficient solution must, therefore, involve a basically engineering approach, but one that is blended with a real understanding of the human side of the problem. There is little value in designing the most finely engineered products that, by their complexity, defeat the handling of the ultimate user, just as the smartest machine is badly designed if it does not lend itself to ready production. To provide the specialised yet balanced attention necessary on the sales side, the need is for designers with qualifications that are of a rather different order from most traditional forms. Besides the psychological, physiological and social understanding of the human customer necessarily required in sound analysis of market requirements, there must be a well developed aesthetic sense in addition to general engineering knowledge. Only if industry can produce designers able to handle these diverse requirements will real efficiency be attainable.

The means at present in use for giving attention to sales points range in all shades from the retention of special "industrial design" consultants to supervision by the chief engineer in the drawing offices and workshops of the firm concerned, the former usually emphasising the concentration on the visual "styling" aspects of sales requirements and the latter the concentration on production and mechanical engineering problems. These tend perhaps to be extremes, and future tendencies must be towards evolving systems that preserve these concentrations while so co-ordinating and extending them that all sides work together to their maximum effectiveness. The solution of the problems involved under sales design will form the principal meeting ground for the designers operating on the "production," "mechanical" and "sales" sides. The success of the co-ordinating efforts here will be the measure of the success of the whole design organisation.

Stated broadly then, the filling of the void produced by the changeover from the sellers' market of past years to the present day buyers' market demands more attention for the sales side of design. This, in turn, demands the employment of specially qualified designers giving full attention to it, and also their effective co-ordination with the production and mechanical engineering sides, the point being that the engineering sides and the sales side are not two independent things that can be considered one after the other, but are two complementary activities which, while using distinct approaches and standards of value, must take cognisance of each other and be worked out together. The problem is one of differentiation and co-ordination. The writer feels that although increased competition has forced Britain into the position of having to extend her methods, the increased opportunities of all-round efficiency can, in the long run, only result in increased feelings of satisfaction amongst all workers in industry and the users of industrial products.

ASSEMBLING METAL PARTS BY SHRINK FITTING

(*Concluded from page* 364)

front of the valve seat locators. The seats are held at the ends of the ducts by movable dogs, which are withdrawn when eight automatic pistons insert the valve seats inside the locators. Piston pressure lasts approximately 5 sec., and close contact between the seat and the locator is ensured by a spring force acting at the end of the run.

Fitting of air-cooled cylinders on aircraft engines: For shrink fitting an interference fit of 0·008 in. was regarded as sufficient. This grip was not achievable by means of liquid nitrogen alone, so that a compound process was used, the engine head being heated up to 360° F while cooling the cylinder by immersion. The cylinder is held on the jig by the adjustable friction of a leather ring and driven by means of teeth on the part and on the jig. The complete operation, including the fitting of the two valve seat inserts and three sparking plug holders, takes approximately 9 min. and requires 3 litres of liquid nitrogen per cylinder.

Fitting of new axle spindles on locomotive crankshafts: The usual expansion process with heating to 750° F cannot be employed for repair because, by heating the flanges, other parts of the axle may move. The solution adopted was to heat the flanges to 212° F and cool the spindle with liquid nitrogen, the grip adopted being 0·022 in. The old spindle was partially machined off, then forced out with a press. For reboring, four distance-pieces were placed and arc-welded on the flanges. The expansion of the bearing was 0·008 in., and the contraction of the axle spindle 0·016 in. Spacers were laid between the flanges, and after grip had started, the distance-pieces were flame-cut to allow free expansion of the axle spindle. Axles reclaimed in this manner have been found to be fully satisfactory.

Reclamation of crankshafts of presses: A two-crank-wrist shaft, 15-18 ft long, was broken in the middle of one of the crank wrists 19·7 in. in diameter. The shaft was repaired by arc welding. The broken parts were fitted together vertically, and held with auxiliary rods arc-welded on the crank. The axle of the crank wrist was then bored at 7·1 in., and a steel core of 7·1 in. diameter, cooled in liquid nitrogen, was lifted with a chain hoist and lowered into the hole, and stopped by a dog at the right position. The break in the wrist was then arc welded, and the auxiliary rods were removed. The dimensions of the boring were first checked by machining a false steel core and passing it in the hole without play. The dimension of the real steel core was only fixed after this test, in order to obtain a grip of 0·008 in.

Fitting of ball bearing rings: The outside rings of steel bearings with a diameter of 5·5 in. were fitted in aluminium wheels, with a grip of 0·0016 in., using a jig with three movable claws.

CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

BEARINGS AND BEARING METALS

Report on Investigations of Thrust Bearings

By L. LELOUP. (From *Revue Universelle des Mines, Belgium*, 9th Series, Vol. 5, No. 8, August, 1949, p. 258-272, 15 illustrations.)

INVESTIGATIONS are made into the friction qualities of thrust washers as used for limited thrusts in conjunction with radial journals. The three types tested are a deep-groove radial ball bearing, 45 × 85 mm dia., a flat thrust washer, 42 × 60 mm dia., made of steel and running against a bronze sleeve, and a wedge surfaced, scraped, steel washer, 42 × 60 mm dia., running against a flat bronze bearing or a white metal bearing.* The thrust bearings are mounted next to a swivel journal bearing and are lubricated from the journal by fixed-ring lubrication. Tests were conducted in a thermostat chamber† held at 29 ± ½° C, and the friction couple was measured by running-down tests with a flywheel mounted on the horizontal bearing shaft outside the thermostat chamber. The lubricating oil used had a viscosity of 5·75 poises at 20° C, and 0·87 poise at 50° C. The load was applied by balanced levers and any radial loading was avoided as far as possible. It was actually some 20 lbs. Although the radial friction cannot be accurately separated from the thrust friction, it only plays a secondary rôle and does not influence the fundamental results. First, two deep-groove ball thrust bearings were used in identical arrangements, one to take up the load from the loading levers, one to transmit it to the bearing, and the friction couples were measured for both of them combined, the ball bearing near the swivel journal then being replaced by the thrust washer to be tested. Test series were taken at 260, 520, and 900 rpm with thrust loads of 0, 55, 110, 165, and 220 lbs.

Results with low thrust loads were rather indeterminate, but definite tendencies could be observed with higher loads. Table I gives the friction couples measured for 220-lb thrust loads with different bearings tested in the set-up described.

TABLE I. FRICTION COUPLES IN LB.-IN. FOR 220 LBS. THRUST.

rpm	Flat steel thrust washer	Steel thrust washer with scraped grooves and wedged faces	Deep-groove ball bearing
260	4·69	2·88	1·09
520	4·35	2·96	1·12
900	6·43	2·95	1·37

The Table shows the substantial advantage that can be gained from a scraped washer as against a flat washer. More than 50 per cent of friction can be saved with a resulting lower temperature of the bearing. The gain increases with speed and load. In special cases, it would pay to fix the best groove and wedge form and use hard steel for wear reduction. (A wedge angle of about 40 minutes was used here.) Oil should be fed in at the inside diameter and, for high speeds, the groove should stop short of the periphery. There is little difference in friction whether a bronze or white metal bearing is used in conjunction with the grooved thrust washer. The ball bearings are again far superior to the scraped thrust washer, over 60 per cent of the friction being saved. It should be remembered, however, that the ball bearing is lubricated by oil from the adjoining journal bearing. If thick grease is used with felt seals, this advantage would almost entirely be wiped out. Results are only valid for thrust bearings in connection with swivel journal bearings. If fixed journals are used, the resulting misalignment of the thrust washers would invalidate all the conclusions drawn from these tests.

*The wedge profiles were designed to allow rotation in either direction.

†See *The Engineers' Digest*, Nov., 1947, p. 385, and June, 1948, p. 189.

ELECTRICAL ENGINEERING

New Method for the Calculation of Commutation Losses of D.C. Machines

By A. G. IOFFE. (From *Vestnik Elektropromyshlennosti, Russia*, Vol. 19, No. 3, March, 1948, pp. 20-21, 1 illustration.)

THE STANDARD methods of calculating commutation losses require the determination of the sinusoidal components of the step-shaped curve representing the current in the armature winding. The object of this article is to present a simplified method which gives directly the increase in resistance and the coil losses due to current reversal.

The step-shaped curve representing the rate of change of the leakage flux, $d\Phi_s/dt$, can be approximated with sufficient accuracy by a sine curve. The leakage flux Φ_s is then also a sine curve. The change of leakage flux Φ_s occurs over a circumference $(u_k - 1 + \gamma)\ \beta_k$, where u_k = number of commutator bars per slot, β_k = commutator pitch, and γ = number of bars covered by the brush.

The second layer in a slot starts commutating earlier or later and is determined by $\pm\ \epsilon_k\beta_k$, where ϵ_k = shortening of the pitch. The current is fully reversed after a time

$$T = 2\,(u_k - 1 + \gamma)\,\beta_k/v_k$$

the corresponding frequency being $f' = 1/T$.

The effective conductor height ξ is determined by

$$\xi = h\sqrt{b_{Cu} f'\lambda/(b_s \times 2500)} \qquad .. \quad (1)$$

where h = actual conductor height, b_{Cu} = width of copper, b_s = width of slot, and λ = conductivity.

The phase-angle between the layers of a slot is

$$\alpha = \pi\epsilon_k/[u_k - 1 + \gamma] \qquad .. \quad (2)$$

and the coefficient determining the increase in current for the n-th layer is given by.

$$k_n = \xi\left\{\varphi + [(I_u^2 + I_n I_u \cos\alpha)\psi/I_n^2]\right\} \qquad (3)$$

where I_n = current in n-th layer, I_u = current between the layer and the bottom of the slot.

For a winding consisting of two layers, we have

$$k_n = \xi\left\{\varphi + [(1 + \cos\alpha)\psi/2]\right\} \qquad .. \quad (4)$$

If we take into account the part of the winding outside the slot, the coefficient is then

$$k_\Phi' = (k_n + e)/(1 + e) \qquad .. \quad (5)$$

e denoting the ratio of the non-active length of the winding to the active length.

The coefficient k_Φ' refers to the increase in resis-

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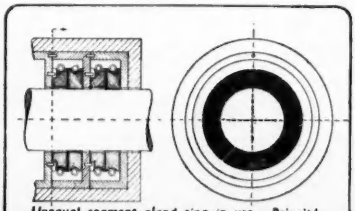

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tance of the short-circuited coils only, and should not be confused with the coefficient k_Φ for the entire armature winding.

The commutation losses are easily determined, when k_Φ' is known, by means of the relation

$$P_k = r_c' \times 2p\,(u_k - 1 + \gamma) \times (I_a/\sqrt{2})^2 \qquad (6)$$

where I_a = current in the parallel branch, and $2p\,(u_k - 1 + \gamma)$ = number of coils being commutated. The factor

$$r_c' = (k_\Phi' - 1)r$$

represents the increase in resistance of a coil corresponding with the commutation losses.

GLASS

Electrically Conductive Glass is Useful for Heating Applications.

(From *Materials & Methods, U.S.A.* Vol. 30, No. 2, August, 1949, pp. 70-71, 2 illustrations.)

Two of the standard insulating materials, glass and rubber, were produced as semi-conductors of electricity during the war, and are now finding civilian uses.

The conductive glass trade-named " Electrapane " is made by the Libbey-Owens-Ford Glass Company, which holds a patent on the method of manufacture. An electrically conductive glass called " Nesa " is also made by the Pittsburgh Plate Glass Company.

Conductive glass can be applied for the following uses :—

1. non-fogging and non-icing glazings for motor cars, ships, locomotives, tractors, and commercial aircraft ;
2. transparent shielding against stray radio frequency radiation, for radio control towers, etc. ;
3. as a heat-generating element, for instance in bath mirrors where the glass serves as a radiator, or as a wall material for tanks in which liquids must be maintained at temperatures somewhat above room temperature.

The " Electrapane " is made electrically conductive by applying a transparent film of stannic oxide, 0·00002 in. thick, on one or both surfaces of the glass. The film is quite durable when exposed to the weather, and is applied at approximately the temperature of the softening point of glass. The residual stresses produced can be the cause of slight optical distortion. The thinnest glass is about 7/64 in. thick, but thicker sheets, having increased mechanical strength due to greater strain hardening, are generally to be preferred.

For non-fogging glazing, the glass is made as a safety laminate, with an intermediate layer of plastic, the conductive film being applied at a temperature slightly below 150° F on one of the inside faces of the glass in contact with the plastic. The glass layers protect the film against short circuits or shocks.

The resistivity ρ of the film is approximately 100 ohms per square area, and the resistance of a panel of width b and height h (h = distance between the end electrodes) is given by

$$R = \rho\, h/b \cong 100\, h/b \text{ [ohms]}$$

The current is supplied by electrodes of copper foil, fused silver flux, or sprayed metal, extending over most of the width of the film at both ends of the path, in order to avoid excessive current densities at any given point. Curved electrodes and irregularly shaped panels produce non-uniform heating, and hot spots are then apt to occur at the ends of the electrodes, so that for the present, the glass is used mainly in flat rectangular shapes which give the best results.

The non-laminated glass can withstand temperatures of up to 450° F without appreciable alteration of its resistivity, and either direct or alternating current can be used. Potentials of up to 440 volts across the film have been successfully tested at low temperature without breakdown. To de-ice aircraft glazing, approximately 3000 BTU or 800 watt-hour is required per square foot of glass. This heat is obtained by applying sufficient voltage, or by arranging a number of intermediate electrodes equidistantly spaced across the path, instead of using one electrode at each end of the path.

MATERIALS TESTING

Appliance for Testing Piston Rings

By P. G. Korolev. (From *Zavodskaya Laboratoria, Russia*, No. 10, 1948, p. 1271, 3 illustrations.)

The appliance has been designed for the testing of locomotive engine piston rings, but can equally well be used for any other piston rings. It enables determination of the elastic and residual deformation of the ring under the influence of concentrated forces, the extent of the latter being measured by the appliance. The schematic drawing in Fig. 1 illustrates the principle of operation.

The deformations can be determined over the entire circumference of the ring, thus showing up any local material defects, due to the presence of which the elastic properties of the material are radically altered.

The force is applied through the upper stop 1 (Fig. 1) by turning handwheel 2. The deformation is indicated by the difference in the movement of the upper and lower stops 1 along the scales 3 and 4.

Elasticity of the ring is determined with the aid of dynamometer 5, also along scale 3.

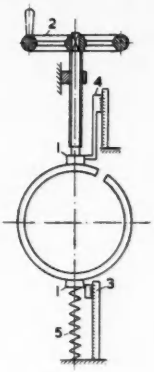

Fig. 1.
Schematic arrangement of test gear.

IN THIS ISSUE—*Concluded from page* 339.

irons. One of the major drawbacks of ordinary cast iron is its notch sensitivity due to graphite inclusions. By hot working the material, the graphite inclusions are given a definite orientation ; in addition, the grain is refined by re-crystallisation and acquires a fibrous structure. The resulting material is, therefore, far superior to ordinary cast iron if properly handled with respect to working orientation. Tensile strengths of 75 tons per sq in. can be reached, with elongations of 5 per cent. These qualities are combined with the known advantages of cast iron, such as damping capacity, wear resistance and ability to absorb and withstand small shocks indefinitely. Since production is not more expensive than that of steel, this material seems to warrant closer investigation into its possible use as a special-quality material.

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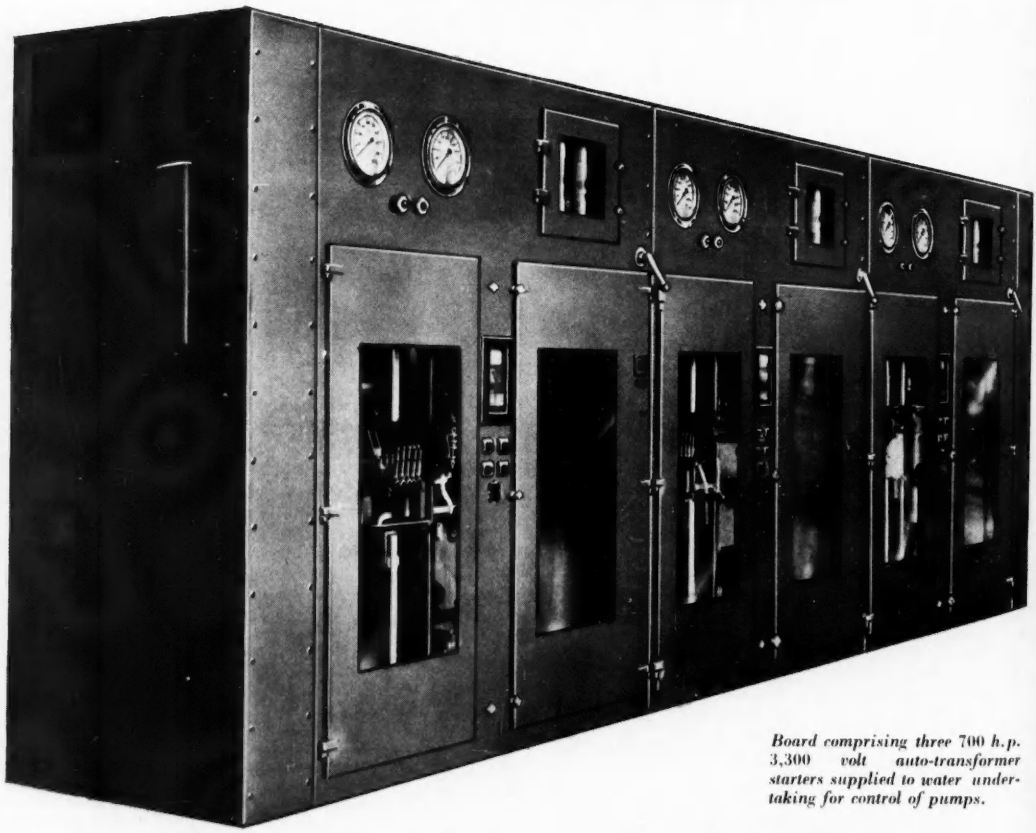

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SURFACE TREATMENT

Electropolishing Brass

By W. G. AXTELL. (From *The Iron Age, U.S.A.*, Vol. 163, No. 26, June 30, 1949, pp. 48-51, 3 illustrations.)

BRASS articles of irregular shape, such as small brass luggage parts, can be given a clean surface, free from abrasive or burnishing particles, by electropolishing, using a new bath and polishing process for which patents are being secured by Shwayder Brothers, Inc., of Denver, Colorado.

It has been found by Jacquet and Walton* that the addition of certain organic compounds increases the polishing effect of the phosphoric acid—water bath used, by promoting anodic film formation. The highly-viscous film formed on the article is thinner on projections extending from the metal surface than in the hollows, so that the metallic salts can be diffused away more rapidly from the peaks.

The bath now used for electropolishing has the following composition :—orthophosphoric acid (85 per cent $H_3PO_4 \cdot H_2O$) 41·5 per cent ; glycerol (1, 2, 3-propanetriol) 24·9 per cent ; ethylene glycol (1, 2-ethanediol) 16·6 per cent ; lactic acid (85 per cent 2-hydroxy propanoic acid) 8·3 per cent ; water 8·7 per cent.

The amount of water added is adjusted so that the specific gravity of the bath is approximately 1·388. To increase surface brightness and the ability to polish around sharply-curved surfaces and surfaces remote from the cathode, other agents, such as glutamic acid and monosodium glutamite are added in minor proportions. (Mannitol, glycine, ethylene diamine, and monoethanol and triethanol amines can also be used.) The brightness decreases with higher concentrations. The brass articles are polished at 78-98° F solution temperature, and at 2 amp. per specimen current.

Film-thickness control, necessary to ensure that the desired brightness is achieved, is obtained by using agitators, or by rotating the wire racks with 60-100 articles slowly, through the bath, to obtain a steady flow past the articles.

Although it is comparatively easy to produce bright surfaces, it is more difficult to obtain surfaces which are free from grain and which have a mirror-like finish. The sharpness of definition or resolving power of polished articles can be evaluated by means of a simple and effective test developed by H. F. Walton*.

A strip of card is ruled with six parallel lines, 1 cm long, 2 mm apart, and each 0·5 mm wide. The operator has the test surface laid before him, and holds the card with its lines on the under side, so that he can see their reflection on the surface of the specimen. The card is then raised away from the specimen until the six lines are no longer distinguishable but appear as a single blur. The height of the strip above the test surface is then read on a centimetre rule held vertically. The tests give reproducible results, with an accuracy of $\pm$ 0·5 cm.

The electrolytic cells are so designed that the articles cannot be stained by accidentally coming in contact with the cathode. This is achieved by making the steel tank the cathode, and inserting in it a ceramic jar 2 in. smaller in diameter. In the space thus formed between the tank and the jar, cooling coils are placed, in order to maintain a constant bath temperature.

After immersion in the bath, the rack is lifted out and allowed to drain for a few seconds, and then goes through a first wash, followed by a second hot rinse at 150° F. The parts are then either dried for storage or varnished and finished directly in the wet condition.

The process has been successfully applied to 70-30 brass, and also to copper alloys such as beryllium copper; nickel silver and cupro-nickel, Monel metal and bronze alloys. Stainless steels and special high-temperature steel alloys, as well as certain aluminium alloys, can also be polished in this manner.

*Paper presented by H. F. Walton before the Division of Physical and Inorganic Chemistry at the 115th National Meeting of the American Chemical Society, San Francisco, March 31, 1949.

Fitting of Rotors on Shafts of Electrical Machines

(*Concluded from page 358*)

Substituting eq. (24) in eq. (11a) gives

$$\epsilon_r = 2 M_h [p_f + Q_h p_e]/(3 E_h) \quad .. \quad (25)$$

and by equating eqs. (25) and (22) we obtain

$$p_e = 2 p_f \Big/ \Big[\Big\{ 3 S_h E_h L/(M_h S_r E_r) \Big\} - 2 Q_h \Big] \; .. \; (26)$$

The value of p_e thus calculated can be substituted in eqs. (10a), (10b), (11a) and (11b).

For very thick hubs ($K_h > 1{\cdot}7$–$1{\cdot}8$), the reaction p_e of the rotor arms is negligible, and we obtain

$$T_{R\,max} = 2M_h (2K_h^2 - 1)\, p_f/3,$$
$$T_{T\,max} = 2M_h N_h p_f,$$
$$T_{R\,min} = 2M_h p_f/3,$$
$$T_{T\,min} = 2M_h p_f.$$

In the case of predominantly tangential stresses,

$$p_f = 3\rho\,(1 - \tau)/2,$$

and for predominantly radial stresses

$$p_f = \theta\mu/(2M_h K_h^2).$$

The above formulae are applicable to most practical cases, and the results are accurate to approximately 2–3 per cent.

In the series of tests carried out by the Società Nazionale Officine Savigliano, the validity of the above formulae was confirmed, and the materials generally used had the following characteristics :—

Material : mild steel Aq. 52 ; ultimate tensile strength 52 kg/mm² ; elongation 6 per cent ; stress at elastic limit : $\theta = 25$ kg/mm² ; $E = 2200$ kg/mm² ; Poisson's ratio : $m = 3{\cdot}6$. All fitted surfaces machined. Eccentricity error less than 0·01 mm. The hubs were shrunk onto the shafts, and no superficial oxidation of the contact surfaces was allowed. The ratio i/D varied between $2{\cdot}1 \times 10^{-4}$ and $5{\cdot}3 \times 10^{-4}$, and the fit pressures varied from 32 to 310 metric tons.

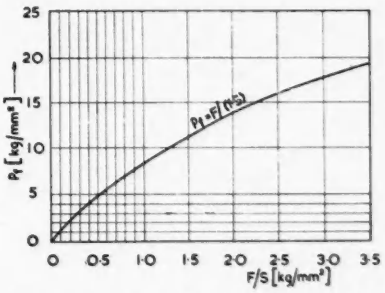

Fig. 2. **Fit pressure p_f as a function of the specific pressure F/S.**

The value of the coefficient of friction $f = F/(SP_f)$ — F being the forcing pressure in tons, and S the total contact area—was determined experimentally on various rotors. A curve used for determining F/S, and hence F, since S is known from the calculated value of p_f, is given in Fig. 2.

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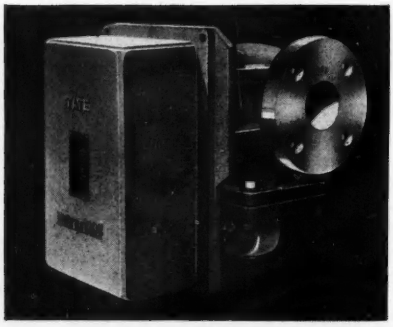

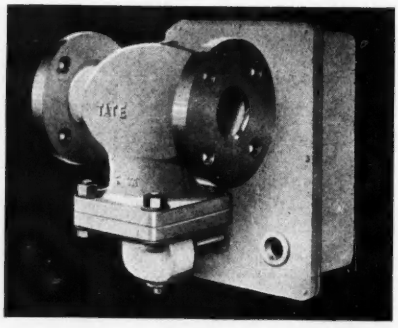

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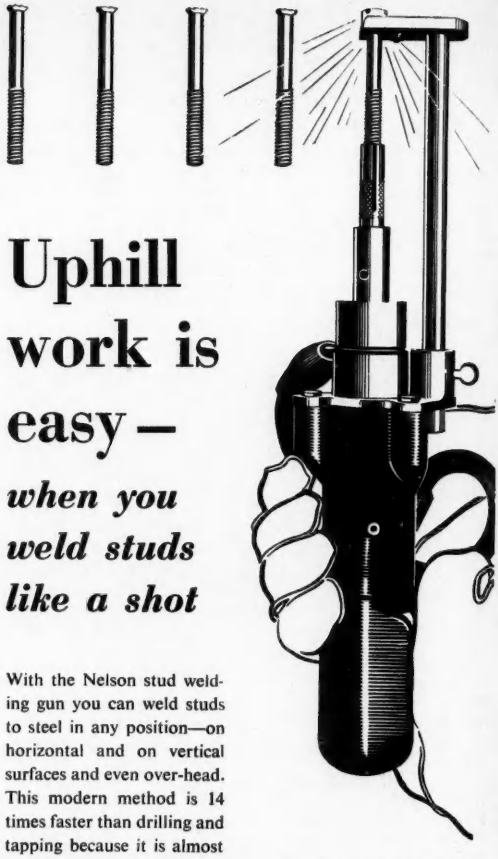

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NEWS OF THE MONTH

PERSONAL

Mr. S. Anderson, B.Sc. (Eng.) (Lond.), has been appointed chief illuminating engineer of The General Electric Co., Ltd., Magnet House, Kingsway, London, W.C.2.

Professor H. E. M. Barlow, B.Sc. (Eng.), Ph.D., M.I.E.E., A.M.I.Mech.E., has been appointed Dean of the Faculty of Engineering at the University College, Gower Street, London, W.C.1.

Mr. H. M. Garner, C.B., has been appointed Chief Scientist in succession to **Sir Ben Lockspeiser.** For the last three years Mr. Garner has been Principal Director of Scientific Research (Air) in the Ministry of Supply.

Mr. K. R. Hopkirk, M.A., chief mechanical engineer of the British Thomson-Houston Co., Ltd., Rugby, has been elected a director of the Company.

Mr. A. V. Lawry, A.M.I.E.E., and **Mr. C. A. Bradbury, A.I.E.E.,** have been elected directors of Contactor Switchgear Ltd., Moorfield Road, Wolverhampton.

Mr. A. J. Murphy, M.Sc., has been appointed Professor of Industrial Metallurgy at the University of Birmingham.

The Rt. Hon. Viscount Portal of Hungerford, K.G., G.C.B., O.M., has joined the board of the Ford Motor Co., Ltd., Dagenham, Essex.

Mr. W. A. Robotham, M.I.Mech.E., chief engineer, Motor Car Division, and **Mr. J. D. Pearson, B.Sc.(Eng.) (Lond.), Wh.Sc., A.M.I.Mech.E.,** deputy general manager, Aero Division, Rolls-Royce Ltd., have been elected directors of the company.

Mr. W. A. Smyth, M.I.Mech.E., M.I.Loco.E., M.I.P.E., M.I.I.A., director and general manager of Henry Meadows Ltd., Wolverhampton, has been elected to the Board of Directors of W. G. Bagnall Ltd., Stafford.

Sir Reginald Stradling, F.R.S., has been appointed Dean of the Military College of Science, Shrivenham, Wiltshire.

Mr. I. Wans, M.I.Mar.E., A.M.I.C.E., A.M.I.Mech.E., has been appointed chief engineer, engine department, and **Mr. G. B. R. Fielden,** chief engineer, turbine department, Ruston & Hornsby Ltd., Lincoln.

Mr. R. Weatherall, A.M.I.Mech.E., has been appointed general manager of Wilton Row Extensions Ltd., 20-21, Essex Street, London, W.C.2.

PROFESSOR F. L. MEYENBERG

We announce with deep regret the death of Professor F. L. Meyenberg, who died suddenly on the 1st October, at the age of 74, in Frankfurt a.M., whilst on a lecture tour in the Western zones of Germany.

Professor Meyenberg has been well known in the engineering world as an expert on production organization, including time study and rate fixing. From 1926 to 1933 he was professor at the University of Brunswick, a position which he resigned when Hitler became "Fuehrer" of the German nation. He came to England where he joined the United Steel Company in Sheffield. Attached to the research department of United Steel, he remained in the services of the company until the end of 1939.

Professor Meyenberg's literary activities include numerous articles published in the leading German technical journals, the Zeitschrift of the V.D.I.; "Werkstattstechnik"; "Technik und Wirtschaft"; "Maschinenbau," etc. His book on "Einfuehrung in die Organization unter Berueckschichtigung der Werkskostenberechnung," published by Julius Springer, has reached three editions.

In England, Professor Meyenberg wrote three books :—"Economic Control of Iron and Steel Works," with a Foreword by Sir William Larke, K.B.E., published in 1942 by Chapman & Hall Ltd.; "Time Study and Rate Fixing," published in 1945 by "The Engineers' Digest" in co-operation with Sir Isaac Pitman & Sons, Ltd.; and "Industrial Administration and Management" to be published posthumously by Sir Isaac Pitman & Sons Ltd., in the near future.

Professor Meyenberg was a member of The Institution of Mechanical Engineers and honorary member of The Institute of Economic Engineering.

In 1943 Professor Meyenberg joined the editorial staff of our journal and he will always be remembered for the part he played in the development of "The Engineers' Digest."

DR. GEORG SCHLESINGER, Hon.M.I.P.E.

The death of Dr. Georg Schlesinger, who died on October 6th, at the age of 75, will prove a serious loss to production engineering.

Dr. Schlesinger, who was an expert of world renown in production research, was Director of the Research Department of the Institution of Production Engineers from its inception in January, 1939, until his retirement in 1944, after which his services were retained as Consultant. Prior to this appointment, he had been responsible for the layout of many modern factories in Germany, Belgium, and Holland, and other countries.

Although in the United Kingdom Dr. Schlesinger was perhaps best known for his work in connection with machine tool tests and alignments, there was no aspect of modern production with which he was not familiar. He was the author of many works published throughout the world, some of which were accepted as standard works of reference.

In recognition of his invaluable services to the Institution, he was elected an Honorary Member in 1942.

BUSINESS NOTES

Burton, Griffiths & Co., Ltd., Mackadown Lane, Marston Green, Birmingham, have been appointed sole agents in Gt. Britain, N. Ireland and The Irish Free State for the "Beco" super production thread rolling machines manufactured by Batchelder Engineering Co., Springfield, Vermont, U.S.A.

The Professional Engineers Appointments Bureau (for Civil, Mechanical and Electrical Engineers) has moved to 9 Victoria Street, London, S.W.1. Telephone: Abbey 1737.

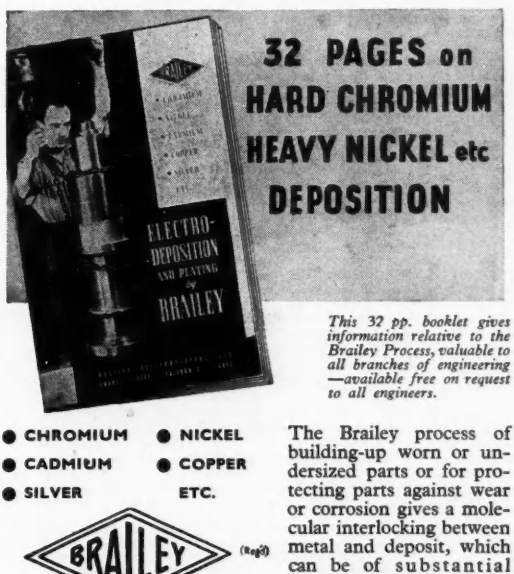
Ericsson & Pye Sign World-Wide Agreement.—Ericsson Telephones Ltd., of London and Beeston, and Pye Ltd., of Cambridge, have signed a working agreement for a world-wide link-up. Besides its domestic radio and television interests Pye Ltd. has export, installation and servicing branches in all countries of the world for communication products, while Ericsson Telephones Ltd. has similar organizations for dealing with telephone, cable and all types of land-line installations.

The new arrangement will allow each Company to handle in future only the type of work in which it specializes, thus ensuring increased efficiency and economies, while through the agreement these two firms will pool their technical, engineering and manufacturing resources as well as their very wide contacts in a serious effort to capture valuable world markets.

STUD WELDING SPEEDS APPLICATION OF ALUMINIUM INDUSTRIAL ROOFING

The Reynolds Metals Co. in America has adopted the Nelson electric-arc stud-welding method for attaching .032-inch industrial corrugated aluminium roofing and siding direct to structural steel framing members.

The method offers important possibilities in reducing application costs, thereby making a more economical construction at the



This 32 pp. booklet gives information relative to the Brailey Process, valuable to all branches of engineering —available free on request to all engineers.

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FUEL RESEARCH STATION

THE importance and wide variety of the work of the Fuel Research Station, of the Department of Scientific and Industrial Research, was described by its Director, Dr. A. Parker, at the conference of the National Smoke Abatement Society in Harrogate.

Coal preparation and cleaning, carbonization and gasification, the efficiency of boilers and furnaces, had been among major questions examined, and research was in progress on the production of oil and chemicals from coal by various means.

Intensive experimental work was in progress to ascertain the possibilities of using powdered coal in place of oil as the source of energy for gas turbines.

The Fuel Research Station had played an important part in the advances in appliances for domestic heating that had taken place during the past few years, and in the production of appliances for preventing or reducing smoke. Fundamental problems of domestic heating and of methods of testing appliances were being studied.

Advances in the measurement of atmospheric pollution, now under the guidance of a committee of the Fuel Research Board, had been made, through work carried out at the Station, and there were now over 500 instruments in use in many parts of the country, with which local authorities are making daily and monthly observations of deposited solid matter, pollution by smoke, and sulphur and the intensity of daylight.

The war work of the Fuel Research Station included the fitting of over 1000 merchant ships with a simple new device for preventing smoke emission from ships in convoy, the development of mobile gas producers for vehicles, flame-throwers, the production of hydrogen for barrage balloons, and research on the recovery of sulphur from pyrites needed urgently for the manufacture of sulphuric acid.

NEW TYPE ATOM SMASHER

STANFORD physicists are ready to begin tests on the first 15-foot section of the university's new type atom smasher—the electron linear accelerator.

The initial section of the accelerator is expected to produce 70 to 80 million electron volts compared to the billion volt output which the physicists hope to have when the atom smasher is completed to its 160-ft length.

This was reported by Richard F. Post of the Stanford physics' department staff at a meeting of the American Institute of Electrical Engineers in San Francisco.

Dr. William W. Hansen, Stanford physicist who died last May directed the design and development of the electron linear accelerator which is now being constructed in the university's new Microwave Laboratory. It is being built under a contract with the Office of Naval Research.

Tests on a working model which produces electrons of more than six million volts have indicated the basic soundness of Dr. Hansen's theories and design.

A high-power vacuum tube has been built and operated at an output adequate to meet the enormous electrical appetite of the completed accelerator. Power for the accelerator will be generated by a series of such tubes which will be turned on in pulses of only a little over a millionth of a second, then turned off for a sixtieth of a second before being turned on again. Electrons will speed along this powerful microwave at a speed close to that of light.

Atomic disintegration occurs when a speeding electron, to which tremendous mass and energy have been imparted by acceleration, strikes an atom which happens to be in its path. The resulting collision is expected to smash large hunks from the atomic nucleus and to produce heavier mesons than has been hitherto possible in laboratories. Mesons are by-products of cosmic ray collisions.

If Stanford physicists are successful in producing billion volt electrons—and it may not be before late in 1951 that they can do so—a number of new experimental fields will be opened in nuclear research.

Assuming that the billion volt electrons will come out of the accelerator "like water out of a spigot," Post indicated that these heavy electrons will provide a new method of studying both the properties of the electron and the proton and possibly the neutron.

The 15-foot electron linear accelerator will be the first one in its energy range in which it will be possible to bring the electrons out of the accelerator into the laboratory for study. This will be true also of the 160-foot model.

In other high-energy electron accelerators, it has not been possible so far to accomplish this feat, which will make it possible for the first time, outside of cosmic ray studies, to investigate adequately the effects of collisions between electrons and nuclei.

It may also be possible to use the electron beam as a sort of super "diffraction camera" to study nuclear charge distributions.

The accelerator will also make possible a careful experimental check of the "shower theory," which is based on the phenomenon that particularly energetic particles produce bursts of atomic bits and radiation when they collide. The existence of atomic "showers" is known from cloud chamber experiments, but no accurate checks of the theory have been made.

The Microwave Laboratory is under the direction of a committee composed of Dr. Edward L. Ginzton, director; Dr. Leonard I. Schiff, head of the physics department; and Dean Frederick E. Terman of the School of Engineering.

Atomic Energy Levels.—A critically evaluated compilation of all known data on the energy levels of elements of atomic number 1 through 23 has recently been published by the National Bureau of Standards and is now available from the U.S. Government Printing Office, Washington, D.C., at a cost of $2.75 a copy. Designed to meet the needs of workers in nuclear and atomic physics, astrophysics, chemistry, and industry, the publication is an up-to-date compendium of all energy levels for these elements exclusive of those due to the hyperfine structure ascribed to atomic nuclei.

The present volume is the first of a series being prepared at the Bureau as part of a general program on atomic energy levels derived from observations of the optical spectra of atoms and ions. Section 1 of this volume (hydrogen through fluorine) has previously been published separately. The new series, represents the cooperative effort of scientists throughout the world and will constitute the first compilation of atomic energy levels in the last 18 years, during which time the number of known energy levels has increased by a factor of 4 or 5.

In this series, spectra are presented in order of increasing atomic number, and under a given atomic number they are listed in order of increasing stages of ionization. For each spectrum a selected bibliography covering the analysis is given. The energy levels are tabulated in the related groups that form spectroscopic terms, counting upward from the lowest as zero. Electron configurations are also given in the tables, together with term intervals, Lande *g*-values, and term designations in a uniform notation. For the more complex spectra, arrays of observed terms and their electron configurations are included. Similar arrays of the terms predicted by theory for important isoelectronic sequences are given in the introduction.

International Technical Committee on Ball and Roller Bearings.—At the recent Conference of the International Organization for Standardization (ISO) in Paris, of which the B.S.I. is the United Kingdom member, the first meeting of the International Technical Committee on Ball and Roller Bearings took place on 27th and 28th June. Delegations from nine nations were present, including the United Kingdom. The Secretariat for this Committee is handled by the Swedish Standards Association.

Dr. H. Tornebohm (Sweden) in an opening historical review of previous international conferences on this subject, recalled that at Helsinki in 1939 it was decided to try to draw up a general plan of boundary dimensions into which ball and roller bearings of all sizes and types could be fitted.

This scheme or framework was now put forward by Sweden. It concerned itself in the main with dimensions affecting the fitting of the bearings, as it was considered useless to create international standards for constructional details which are in continuous development, and it was extended to include the recently developed miniature bearings. This scheme was embodied in a document which had been originally circulated four months before the meeting. The Americans, however, had since circulated a document showing numerous differences both from the dimensions now proposed, and from the recommendations incorporated in the scheme. To meet these as far as possible, in view of the very large American production of bearings, the secretariat submitted at this meeting a revised scheme, but the time was too short for this to be studied, and little progress could therefore be made.

Considerable attention was, however, given to certain bearing sizes produced in considerable numbers in the United Kingdom and in the U.S.A. to dimensions different from the recommendations particularly the bearings of bore 95 and 100 mm, respectively, in the metric heavy series. The British delegation proposed that the outside diameters of these bearings be altered to 250 and 265 mm., respectively, in accordance with long established British practice. No agreement was reached, however, the Continental countries opposing this on the grounds that these dimensions had been altered at earlier conferences to 240 and 250 mm., in order to fit in better with the general plan.

Amongst other details discussed were the 10 mm. ball bearings which have been manufactured in large numbers with a width of 14·3 mm. in the U.S.A. instead of 14 mm. The Swiss delegation said that this had caused difficulty in interchangeability, but the American delegation held out no hope of their industry being prepared to abandon the greater width. It was generally agreed that further study would be required before the Swedish revised proposals could be profitably discussed, and a sub-committee was appointed for this purpose with Switzerland as secretariat.

The subject of taper roller bearings was introduced and the American delegation explained that these were not produced in the U.S.A. to a boundary dimension plan as was suggested in the Swedish proposals, but were designed to meet the specific needs of users. A sub-committee representing six nations with U.S.A. as secretariat and including the United Kingdom, was appointed to go into the whole question.

Establishment and Maintenance of the Electrical Units.—A description of the new system of electrical measurement using "absolute" units, adopted by the International Conference of Weights and Measures and officially instituted January 1, 1948, is described in this new booklet.

The purpose of the booklet is to record an account of the working of the international system of electrical units, with particular reference to the maintenance of the international units in the 37 years preceding the adoption of the new units; to point out the trends of development that made them obsolete; and to record the official steps by which they were superseded. Also described are the methods used in the measurements that now form the basis for the

new absolute units, in which all certifications for standards and instruments are now given by the National Bureau of Standards. In addition, United States laws, important conferences and resolutions pertaining to the adoption of the new units are reproduced in the appendices of the booklet.

Circular C475 *Establishment and Maintenance of the Electrical Units*, by F. B. Silsbee, 38 large double-column pages, illustrated with 23 diagrams, half-tones, and tables, 25 cents a copy, is available from the Superintendent of Documents, U.S. Government Printing Office, Washington 25, D.C.

Charts for Orthographic Isometric and Perspective Drawings. Mr. E. H. Hassler, M.I.Mech.E., has developed a series of charts, which, inserted under tracing paper, greatly facilitate ortographic, isometric and perspective drawings. These drawing guides are now manufactured by the Ozalid Company, Ltd., 62 London Wall, London, W.C.2, under the fitting name "Magic Carpets." Leaflets describing the method and showing examples of the work which so easily can be accomplished by taking advantage of the practical drawing guides, have now been published and can be obtained free on application to the Ozalid Company.

New No-Pressure Plastics Moulding Process. One of the biggest obstacles in the use of high-strength reinforced plastics as engineering materials has been the pressure required to make mouldings with satisfactory strength properties. This consideration has been particularly important in the aircraft and ship-building fields, where a moulding process which would produce a smooth, rivetless contour holds very great promise.

The first publication of a no-pressure moulding process devised by the Royal Aircraft Establishment, Farnborough, is described in "British Plastics," October issue. Using this technique with a material such as Durestos, it is possible to produce large and complex mouldings at low cost. The process will find application in many industries. Examples are spars, ribs and internal bulkheads for aircraft, dinghies and small yachts to give smooth and extremely strong hulls, the building of prototype in engineering practice, experimental coachwork in the motor industry, etc.

Low-Cost High-Speed Wind Tunnel. A serious bottleneck which threatened to impede the progress of British aeronautical research into the problems of supersonic flight has beeen cleared by the English Electric Company which, by devising a new principle, has made possible the construction of high speed wind tunnels at a cost far below that of conventional tunnels.

Most of the wind tunnels used by private aircraft manufacturers are comparatively small and limited to test work at relative speeds well below those which are now becoming general in flight. The majority are built on the closed-circuit principle, in which air is constantly circulated in an endless tunnel by an electrically-driven fan.

Models and parts of aeroplanes and engines are placed in a "working section" of the tunnel which is normally situated at a point a short distance upstream from the fan, so that air is drawn, rather than blown, over the model under test. In this way the date obtained is more accurate.

The power needed to circulate the air depends upon the size of the wind tunnel and the speed of the airflow required. Some tunnels have motors of only 200 hp, but one of the largest in this country—it belongs to a Government research establishment and cost hundreds of thousands of pounds to build—needs a motor of 2,000 hp.

The Germans evolved and put to use a high-speed wind tunnel of unconventional design. This incorporated a huge "sphere"—it had a capacity of 26,500 cubic feet—from which air was pumped until a state of almost complete vacuum was reached. On the turn of a quick-release valve, air was drawn in at a rate from three to four times the speed of sound through a tunnel which contained the working section of the apparatus.

This rate of flow could be maintained for only 15 seconds, after which it fell off rapidly, the sphere taking less than one minute to regain normal atmospheric pressure. The working period was, therefore, extremely short.

This wind tunnel was dismantled and re-erected in the U.S.A. after the war. The original sphere was replaced by one of 52 feet diameter and 73,600 feet cupic capacity. On 24th November, 1948, a rate of flow of 1,575 mph through the working section was recorded. Because of the low temperature created by the high speed of the airstream this corresponded to 3,960 mph.

As an indication of the size of the German tunnel, the huge sphere occupies less than a quarter of the total space required by the complete installation. Three powerful electrically-driven pumps take 15 minutes to empty the sphere.

In England, the cost of building a high-speed wind tunnel of conventional design was too high for any individual manufacturer to face, and the only tunnel capable of testing at high subsonic speeds was one built and operated by a Government research establishment. This has a motor of 4,000 hp.

An increasing need for high-speed research, and the shortage of available facilities, conspired to create a serious situation, and there was a danger that the capacity of this single unit would be overtaxed, and British progress gravely delayed. The English Electric Company had already designed and built its own conventional wind tunnel when the need for higher speeds became apparent. It was then that the company's engineers conceived the idea of applying a new principle to wind tunnel work and starting a new process of development.

They discarded the closed-circuit principle and turned to the ejector (or straight-through) system, in which air is taken from atmosphere, passed through the tunnel, and returned to atmosphere. To create the airflow they hit upon the simple but ingenious idea of using a turbojet engine as an ejector pump. Their choice fell upon the Rolls-Royce *Nene*, which has a static thrust of 5,000 lb.

The *Nene* is housed in a nacelle carried in a bulge in the tunnel. A space is left between the inner nacelle and the outer wall through which two-thirds of the tunnel air is drawn by the ejector action of the jet, and the remaining one-third direct into the engine.

The working section lies a few feet upstream from the engine nacelle, and before entering the tunnel the incoming air passes through a fine-screen cage which prevents any solid substance from entering the tunnel and damaging either the object under test or the engine.

When no model or other object is in the working section a rate of airflow of nine-tenths the speed of sound (or, in technical language, a Mach number of 0·9) is reached; with a test piece in position, the speed drops to a Mach number of 0·85. The English Electric Company is now working on a wind tunnel to give speeds above that of sound, using no more power than that available from a *Nene* engine.

As to duration, the tunnel could be run non-stop as long as the engine keeps going, without drawing upon external source of power.

A straight-through tunnel of this type would suffer from condensation trouble on most days of the year if the humidity were not lowered at the intake. To prevent this trouble from developing a part of the hot exhaust is collected and mixed with the incoming air. Although this adds moisture, the humidity is lowered by a rise in temperature.

Another advantage of the English Electric tunnel is its compactness. The first has been fitted into a war-time piston-engine test bed, measuring only 60 feet by 34 feet by 18 feet high. From this building extends a large-diameter 40 yds. long diffuser which carries away the ejected air.

The entire installation, including all the accessories and instruments that go with it, costs only a few thousand pounds. As a result of this brilliant stroke of genius, almost every aircraft and aero-engine firm in the country can, if it wishes, equip itself with this vital instrument of supersonic research.

Two companies are already adopting the English Electric Company's design and no doubt others will follow if they find that their lack of a high-speed wind tunnel is impairing their efficiency or leaving a gap in their knowledge.

Classified Advertisements.

The Rate for all classified advertisements is 6d. per word; in bold print 9d. per word; minimum order 6s. Box number advertisements 1s. extra. Instructions together with remittance must be received not later than the 3rd of each month for advertisements to appear in the same month's issue.

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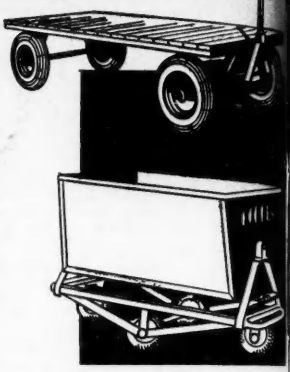

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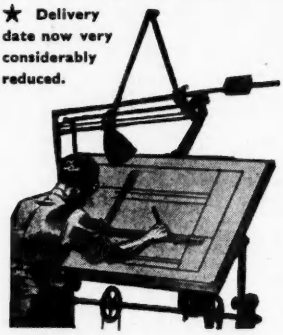

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